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# Heating · Piping and Air Conditioning

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Large Buildings

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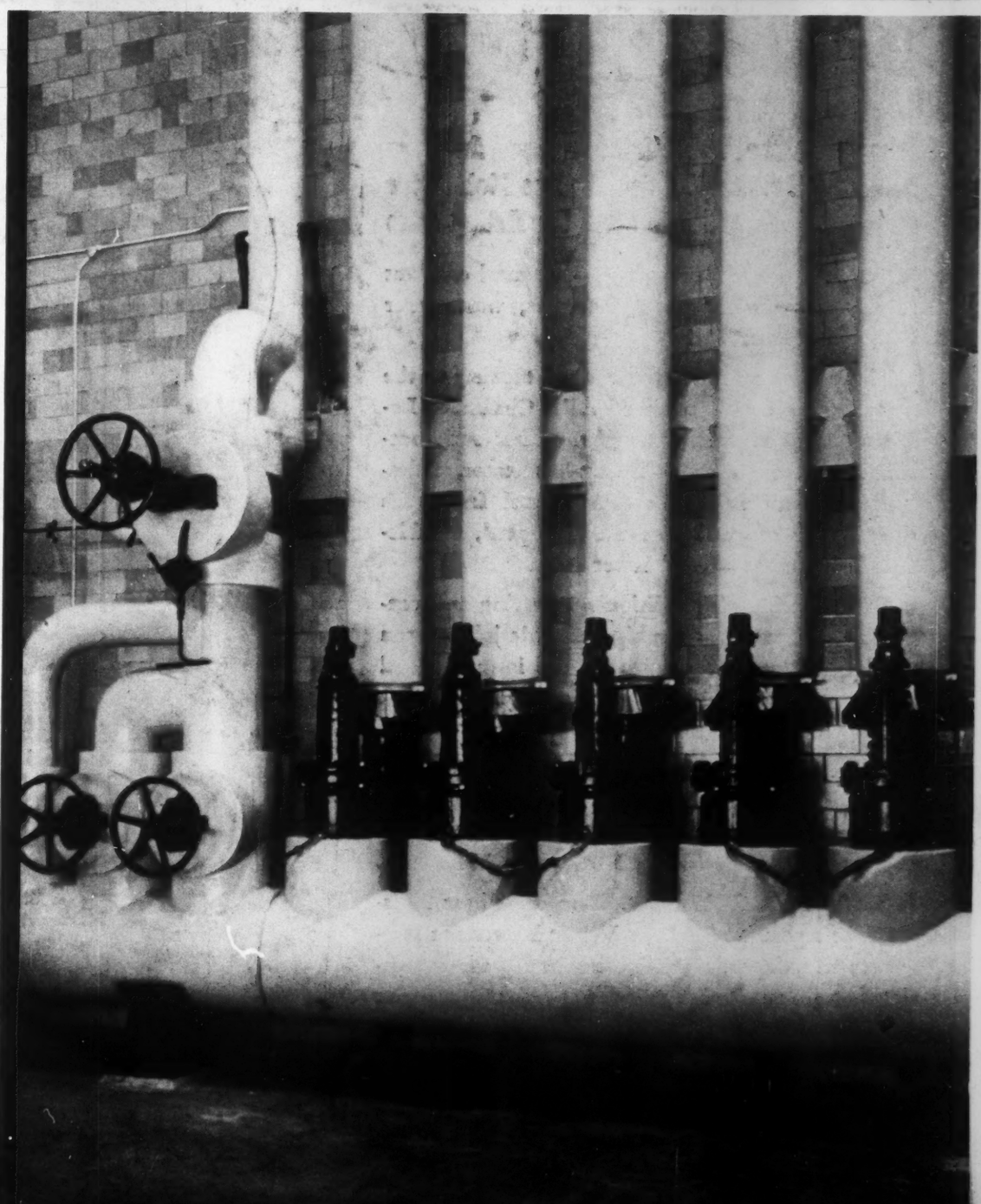
And there are many others



Page 639
Journal Section
The American
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# Quality to the Core

## ...LL-DUTY HORSEPOWER

### Jobs Run Smooth and Trouble-Free When You Use Allis-Chalmers Lo-Maintenance Motors for Air-Conditioning, Heating And Ventilating Units

JOBS THAT keep acting up — demand constant servicing — annoy your customers, waste your time and your profits!

That's why careful engineers are turning to the full-measure construction of Allis-Chalmers Lo-Maintenance Motors. They *know* that such outstanding features as distortion-less stator, indestructible rotor and high carbon steel frame are their surest guarantee of uninterrupted, trouble-free horsepower — and lower costs.

For powering air-conditioning, heating and ventilating equipment — on big and little jobs — you need the drastic economies — the long life and protection these quality motors supply.

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V-BELT DRIVES TOO! You can't beat Allis-Chalmers genuine Texrope V-Belt Drives for low-cost, silent, dependable, high-efficiency power transmission. Ask to see the new Super-7 Texrope V-Belt — far stronger for longer life, longer wear. A 1447

### CHECK THESE FULL-MEASURE FEATURES!

1 **QUIET!** Fan blades and end rings are cast in one piece. Liberal air passages control velocity of entering air — reduce noise to an operating minimum.

2 **PROTECTION!** Stator is treated with special insulating varnish for resistance to moisture, mild acids and age. Shock-proof, cast steel frame with integral feet reduces vibration.

3 **ECONOMY!** Dust-proof, leak-proof bearings save oil, cut maintenance. Special bearing seal eliminates replacement costs.

4 **PERFORMANCE!** Every Lo-Maintenance Motor fully meets every standard of ratings, temperature rise, torques and other characteristics adopted by the NEMA and AIEE. Case histories prove it can't be matched for longer, cheaper, more dependable service!

## ALLIS-CHALMERS LO-MAINTENANCE MOTORS

A COMPLETE LINE FOR EVERY PURPOSE...ONE-HALF HORSEPOWER AND UP

# The Editor's Pages

## INDUSTRIALIST LOOKS AHEAD

Return of peace will confront industrial management and labor with the problem of working out an "orderly retreat" from the present war boom economy, according to W. G. Marshall speaking at the annual convention of the Pennsylvania Electric Association held at Bedford Springs, Pa.

Mr. Marshall, vice-president in charge of industrial relations at the Westinghouse Electric and Mfg. Co., declared:

"The heart of the whole problem is to work out an orderly retreat through collective bargaining. If this can be done in patience, out of it will come an understanding of industrial management on the part of labor that will mean that the full course has been run, that labor has come of age, and is prepared for some of the responsibilities it has sought, but for which it was not yet ready.

"While defense activity will continue even after the war beyond anything we knew prior to the present phase, the steam will be off; more leisurely production will replace the feverish haste that now dominates American industry.

"There will be curtailments of employee enrollment, and doubtless these curtailments will come just at a time when a million or two million or more vigorous young men are returning from service. These boys have been given more than moral assurances of the jobs they left to enter training.

"Those who have come into industry from the farm and elsewhere to replace those in service will not unhesitatingly give up their places to those returning home and, on the other hand, management will be desirous to retain those now trained to efficiency. The intense feeling of patriotism will be subsiding. Then, too, the boys returning are not the same who went out. They have widened their experiences, but their special skills have become dulled; they have changed their outlook; and a certain restlessness must be patiently accepted. In all this readjustment of personnel in a declining economy, organized labor is entitled to express its point of view, perhaps pressing for application of the principle of strict seniority.

"Married women will be among the first to be furloughed in the retrenchment program. The marginal groups and the handicapped will be again unemployed. The retirement age will again begin to fall, and boys and girls just out of school will have difficulty in finding jobs.

"But, even preceding reductions in working force, the first shock of retrenchment will be taken up by reducing the hours of work. The work week is already 48 hours in many production locations, and with two or three shifts in effect. Overtime will first disappear, and below that operations perhaps will go to 35, 30 and even lower before vast reductions in force occur.

"Will government take the initiative or will management? It remains to be seen how organized labor, which will have a very real part in the decisions that are made, will react to the application of 'share the work' as a policy. The problem of working out this part of the orderly retreat through collective bargaining will represent a major undertaking for industrial management."

## EMERGENCY STEEL SPECS STUDIED

Increased production of steel from existing facilities during the national emergency by concentration on a minimum number of steel specifications, compositions, sizes and shapes is expected to result from a new project launched by the office of production management.

At the request of OPM, three national organizations—the American Society for Testing Materials, Society of Automotive Engineers, and the American Iron and Steel Institute—will carry out the project with the collaboration of the war and navy departments, under the general supervision of OPM. An administrative committee to direct the work has been formed of representatives of these five bodies with advisors from other interested organizations.

The goal as defined by the administrative committee at its first meeting on September 12 is to establish as promptly as possible a selected list of steel specifications, to be designated national emergency steel specifications, which in effect involves the selection of the minimum number of steel specifications, compositions and sections necessary to meet the requirements of national defense, both direct and indirect.

It is believed that the productive capacity of the steel industry, and of the manufacturing industries using steel for defense equipment, can be materially increased with present facilities by concentration of production upon a reduced number of steels, particularly with respect to alloy steels. Consideration will necessarily be given to non-defense requirements for steel in establishing the list of national emergency steel specifications.

It is the intention of OPM, through its iron and steel section, to use the list as an aid in administering steel priorities and allocations. The administrative committee indicated that the purpose is not to write new specifications, but primarily to select from existing specifications the practical minimum.

---

Otto W. Armspach announced last month the opening of an office for the practice of mechanical engineering at 221 N. La Salle St., Chicago. He will specialize in air conditioning, ventilating and heating.

# LOOKING FORWARD

**By JOHN HOWATT**
**Chief Engineer, Chicago Board of Education**
**Member of HPAC's Board of Consulting and Contributing Editors**

No one would begin a long auto journey without a road map. We first study the map before starting the trip to select our route to determine where we will travel, and in our mind set up a tentative stopping place at noon and a stopping place at night. Then at the end of the day we again study the map and analyze our position, decide what progress we are making, whether or not we are following the right trails, and plan further for the journey ahead.

A blind man will occasionally hit the target but his performance will be haphazard and there will be more misses than hits. Direction and continuity of effort are the most important factors in determining destiny. But it is not enough to have an objective; the objective itself must be worthwhile so when we achieve our purpose it will prove to have been worth the effort.

The proper evaluation of different efforts in life is one of its most difficult problems. The importance placed on different things changes very materially with the number of years we have lived if we develop in a normal way. Matters that take first rank in importance with those at the age of 20 seem trifling to those at 45 —and there is still further readjustment in these values beyond 45. Relative values change, too, with changing conditions and circumstances. So the problem of determining what things in life are important at any one time takes on additional complexities because of the changing conceptions of value.

It has been said that experience is the best school, but the fees are heavy. Without memory man is an imbecile, and memory and history are much the same. We should remember what has gone in the past that it may serve as a guide under similar conditions when repeated in the future. We may guide our feet by the lamp of experience if we move in its light. The wa-hoo bird flew backward so he could view the territory over which he had passed. But this fabled bird was interested only in experience without any application to his future. There may be advantages in this mode of locomotion, but to those who are truly interested in progress their concern must lie more in what is in the future than what is in the past—the past is then of value only in order that we may use it as a guide for our future.

Constructive, sensible planning takes into consideration all the available knowledge on the subject in order that we will not be firing blindfolded at the target. A successful army in the field today credits its victories to the careful planning of each move, each campaign, and that planning begins with the training of the morale and physique of the individual. The same technique will win in personal affairs. No one can guarantee success in any venture, but it is certain that planning followed by continued effort toward the desired objective will improve its probabilities.

Our Nation today is engaged in the greatest effort in preparedness, as measured in money costs, that it has ever undertaken. This effort to build national defense is being carried through after study and planning to determine what is most necessary and the best way to arrive at the objective. The exigencies of the hour make speed one of the requisites, so sufficient time for orderly planning could not be had. This haste made necessary many changes and adjustments as the program developed and perhaps is resulting in the performance seeming out of proportion to the cost. But a proper valuation of our objectives—the preservation of justice, freedom and liberty—will make the cost appear secondary. The program has direction, continuity of effort and worthwhile objectives. It will succeed.

## AID EX-SOLDIERS TO FIND JOBS

Announcing that the selective service system will give the same assistance in finding jobs for men completing their terms of service in the regular army, navy or marine corps as it does to its own selectees and members of the national guard, Brigadier-General Lewis B. Hershey, director of selective service, urged last month the patriotic coöperation of all employers to see that every returning soldier is given prompt civilian employment.

Pointing out that the selective service system has put into operation a carefully studied and organized reëmployment program that functions primarily through its local boards, General Hershey stressed that its successful operation also depends largely upon the full coöperation of the nation's industrialists and other employers.

## PROPOSES TIPPING FOR NATIONAL DEFENSE

Recommendations have been made by the defense coöperation committee of the National Mineral Wool Association to have the association's field engineers and all traveling salesmen of member companies stop using coins for tipping and, instead, give U. S. defense saving stamps for services rendered them.

Books of stamps, which come in 10 and 25c denominations and thus make possible tips of 10, 20 and 25c and other normal amounts, are to be furnished the engineers and salesmen in proper quantity before the start of their trips, according to the plan.

The use of defense saving stamps as tips, NMWA Secretary Wharton Clay explains, is aimed to promote national unity. "The larger the proportion of our population owning defense savings stamps, the more our people as a whole, will feel that they have a personal participation in the nation's defense effort. And, in turn, the more interest they will have in seeing these efforts carried through without delay or interruptions."

## 60 TO 80 PER CENT OF FAN PRODUCTS FOR DEFENSE

From 60 to 80 per cent of the products manufactured by member companies of the National Association of Fan Manufacturers are now being furnished for defense projects, according to an NAFM statement. The application of fans and blowers for these purposes ranges from small ventilating blowers to protect an individual welder from toxic fumes to huge blowers handling 100,000 cfm of air.

**Famous Last Lines** — "By the way, let me see. What is your priority?"

# Heating · Piping and Air Conditioning

*with which is merged the "Aerologist"*

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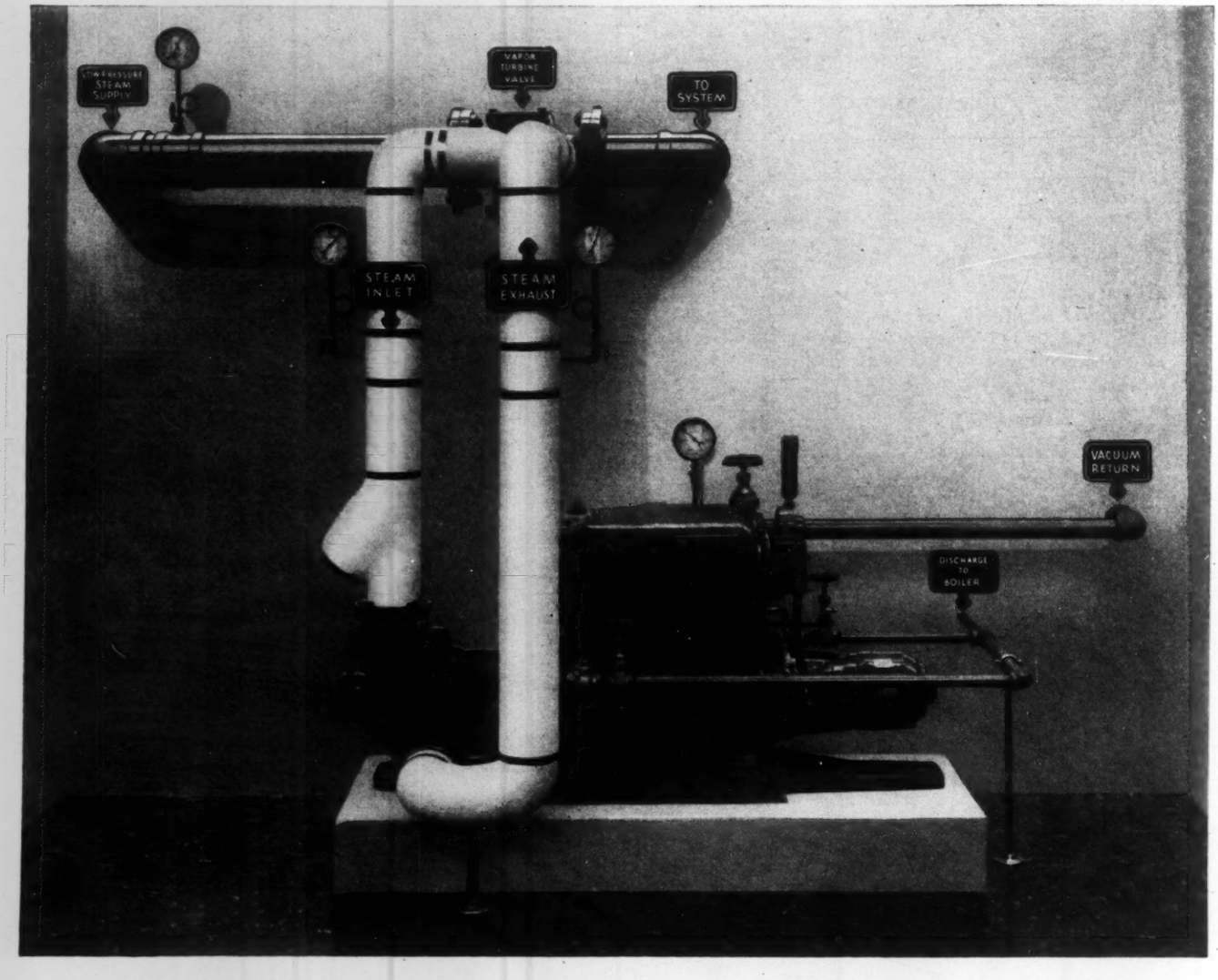
# Where Heat Must Not Fail— Install this SAFE Heating Pump!




In Hospitals, Greenhouses, Schools, Public Buildings, Theatres, wherever heating systems must not fail, install the Nash Vapor Turbine, for it is independent of electric current failure, and continues to operate as long as there is steam in the system.

This is because the prime motive power of this economical pump is a special steam turbine, controlled by a unique "Vapor Turbine Valve", which automatically bypasses from the heating main a small portion of steam, the exact amount necessary to develop the power needed to remove the condensate and maintain the required vacuum on the system. Even this small amount is passed immediately back to the mains, and goes on to the system with little heat loss. This pump operates on any system, high or low pressure.

The Vapor Turbine is a most economical pump, for the elimination of electric current does away with current cost, the largest single item in the operation of an ordinary return line heating pump. Bulletin on request.

## THE NASH ENGINEERING COMPANY

205-A WILSON ROAD, SOUTH NORWALK, CONNECTICUT, U. S. A.

# Production for National Defense

# Two Years of Air Conditioning at the Allison Aircraft Engine Plant

## R. D. Tutt, in Charge of Air Conditioning at Allison Division of General Motors Corporation, Reports on Operation and Maintenance

A REVIEW of the air conditioning installation at the huge windowless blackout plant of the Allison Div., General Motors Corp., at Speedway, Ind., after approximately two years' operation affords an excellent example of the benefits to be derived from air conditioning in defense or other manufacturing plants where precision work is a prime requisite. The control of dry bulb temperature and wet bulb conditions is especially important to precision gages and highly machined parts.

The Allison plant No. 3, which houses the manufacturing and assembly operations producing aircraft engines, has expanded to many times its original scope in the two years it has been operating. As nearly as possible, this expansion in air conditioning has followed the lines of the original installation, which was of approximately 800 tons capacity. It now has a peak

**PRODUCTION for national defense is being immeasurably facilitated by the complete air conditioning of many of our huge new plants; for the windowless, blackout plant, such as that of the Allison Div., control of air conditions is of course a necessity, both for manufacturing operations, and comfort and efficiency of the personnel. . . . Mr. Tutt tells here of the first two years of experience with this air conditioning installation, which has grown in capacity almost since its installation. The original equipment in plant No. 3 had a capacity of about 800 tons of refrigeration; the peak demand is now 3250 tons and there is some 850,000 sq ft of space conditioned. . . . Mr. Tutt tells how constant temperatures—vital to the accuracy of gages—are maintained and how they are varied seasonally. He also discusses the problems presented by the industrial exhaust systems in relation to air conditioning economy, and how they are handled. A thorough description of the ventilation of heat treat areas is included; ventilation air is introduced through orifice holes in the floor around and adjacent to points where the operators of the equipment are situated . . . Mr. Tutt concludes with data on operation, maintenance and costs . . . The management feels that air conditioning is essential in the manufacture of the Allison aircraft engine by maintaining closer manufacturing tolerances and bettering employee efficiency**

demand of 3250 tons, handles 1,100,000 cfm, and has grown from 260,000 sq ft of conditioned space to about 850,000 sq ft.

The buildings of the Allison group are of standard construction, cinder block with brick facing; the roofs contain a 2 in. insulation on steel decks, with a heavy composition roofing laid on the insulation.

The air conditioning system in plant No. 3 is a chilled water installation; 2550 tons of mechanical refrigeration and 700 tons of well water prechilling are used to secure the total tonnage. The chilled water is circulated to a total of 31 separate air conditioning systems, housed in 19 separate penthouses, feeding all sections of the building.

Three 650 ton centrifugal "Freon" compressors, with 600 hp synchronous motors directly connected to each, provide the refrigerating effect in conjunction with the two 300 hp reciprocating compressors of the original installation. Heat exchangers for use with 58 F well water are used first to prechill the returning chilled water, which, after passing through the heat exchangers, is pumped to the compressors and chillers for further temperature reduction, until a final chilled water temperature of 49 F is obtained. Eight centrifugal pumps, capable of handling 650 gpm each, carry the chilled water to the various air handling systems in the penthouses.

Air handling units are fabricated conditioners of various sizes adequate for the respective parts of the building they serve. These conditioners and their insulated cabinets house a set of maximum-minimum dampers, interconnected with the return air louvers from which the air is taken into a capillary air washer. These washers are used in winter for humidification and in summer for air cleaning and dehumidifying. The washers pass their spray water through small heat exchangers, whose exchange water is taken from the main chilled water or hot water line, depending upon the season.

A few of the units first installed depend solely upon these capillary washers for air filtering, but in the main, electrostatic type air filters are used. Because clean air is of prime importance in a manufacturing plant of this type, constant tests are run on the filtering systems in the conditioners, since a comparatively small amount of dirt circulated can affect costly products and machinery.

Water coils of the return bend type are used for final cooling and dehumidifying, and also for heating with hot water in the winter, except in those sections of the building where process steam is available at comparatively low cost, in which case separate steam coils are used for heating.

Heavy duty fans are used to circulate conditioned air in angle iron bound ductwork, which uses high velocity, aspirating type air diffusers for air distribution, except in the original section of the plant, where deflectional grilles are used. The spaces which these systems supply are used as a plenum for the return air, which is finally returned to the units through a common grille situated in the penthouse floor.

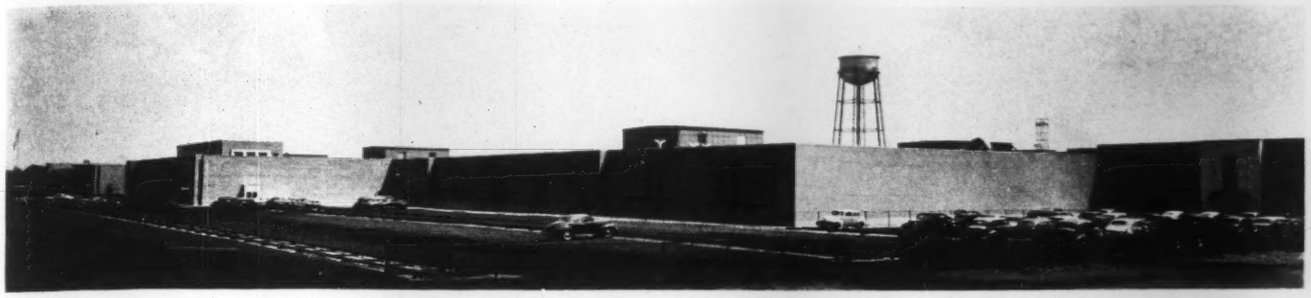
Air controls are used to maintain temperature differential, incorporating control of the outside air, return air, and chilled or hot water supply by means of air operated valves on these lines. It is interesting to note that since each unit serves a separate zone of the building, the individual units are not zoned in themselves, but merely vary their entire capacities as the sun load, or the occupancy load, shifts.

**Interior view of plant No. 3, showing ducts**

## Space Temperatures

The need for constant temperature necessitates a careful survey of daily weather expectations, since if any large increase or decrease in outside temperature is expected to last for any period of time, the building thermostats are set up or down to compensate. This control setting must be done very gradually over a period of several hours, since any sudden change in space temperature will throw close manufacturing tolerances off standard.

The normal design temperature for the plant is 68 F in winter and 78 F in summer. These tempera-



A northwest view of plant No. 3

tures were decided upon as being the most economical to maintain for human comfort and manufacturing tolerances. During permanent seasonal change-over, manufacturing gages must be calibrated for the temperature at which the building will be maintained. Since, in summer, temperature differentials between inside and outside are sometimes severe, the branches of units serving the locker rooms and exit lanes generally, are slightly raised in temperature to allow the workmen to "warm up" gradually before encountering the extremely high outside temperatures. Two degree variation from the design temperatures is the maximum allowed, and the installations are maintained within this limit.

## Plant Hospital

The Allison hospital, which is situated inside the plant, is very modern and is well equipped to handle anything from first aid to a minor operation. For this reason, controls are installed on the air conditioning unit supplying the hospital which will allow the introduction of all outside air when an operation, or treatment requiring the use of ether or other semi-explosive anesthetics, is in progress. An exhaust system connected to the ceiling air outlets (which outlets incorporate a return air chamber in the center) discharges all return air to the atmosphere during such times.

## Industrial Exhaust

Since all of the circulated air in the manufacturing section of the building is conditioned, industrial exhaust systems are proving to be one of the most serious problems to be overcome in the efficient operation of the air conditioning plant.

In one section of the building housing burring operations on magnesium castings, it is, of course necessary to exhaust the magnesium dust. A total of 416,000 cfm conditioned air is being supplied to the section, 25 per cent of which is outside air. The exhaust systems are taking 200,000 cfm to atmosphere. A large saving is accomplished by taking air from the wet cyclonic arresters used to collect the dust, for use in cooling a paint oven conveyor system and the further use of this same air to supply the paint room spray booths. These operations would take 75,000 cfm additional conditioned air from the manufacturing space; hence this amount of air is saved by a relatively small amount of ductwork necessary to carry exhaust air through several operations before throwing the air away.

## Heat Treat Area

This method of handling exhaust problems is used extensively in reducing exhausting of conditioned air to a minimum. An interesting variation of it is used to ventilate the heat treat areas. The newest heat treat is housed in a space 160 ft by 50 ft with a truss line 15 ft from the floor, and is situated in the center of one of the new additions. A 70,000 cfm blower of industrial type is used to supply all outside air to the heat treat from a penthouse intake through a bank of innertube, nonfreeze type steam coils for heating or tempering the air in winter, or as a straight ventilation system in summer.

The outside air is taken through a system of distributing ducts to a 9 ft basement under the heat treat, where it is released to use the basement as a plenum through orifice holes in the floor around and adjacent to points at which the operators of the heat treat equipment are situated. The ventilation air is then taken out of the heat treat by means of mechanical roof ventilators placed directly above the orifice holes in the floor. The capacities of the ventilators vary from 3500 to 8000 cfm, depending upon the amount of ventilation necessary in the section.

The orifice holes were sized to maintain a uniform pressure in the basement section so that no particular area would suffer because of a larger demand elsewhere in the heat treat. In a heat treat of this size, where temperatures of 115 to 130 F are common, a reduction of 12 to 16 F was obtained when the outside temperature was 90. Of course, the actual amount of temperature reduction is strictly dependent upon outside temperatures; the lower the outside temperatures, the more nearly the inside temperature of the heat treat approaches that of the manufacturing section.

Bypass dampers are used around the steam coils for further temperature regulation in the winter. The roof ventilators are capable of handling a total of 75,000 cfm and, incidentally, the intakes to these ventilators are connected to sheet metal stubs which are run to the truss line from the roof line of each ventilator, so that the actual amount of space handled by the system is only that from the truss line to the floor. This arrangement resulted in the initial cost of equipment being materially reduced, and still provided approximately 34 air changes per hour in the space. Since a balanced pressure thus exists, very little air is robbed from the main conditioning system to supply the section.

The problem of isolating the degreasing and parts washing operations, which added to the moisture content of the conditioned air and caused oil odor circulation from one system to the other, was also solved

by housing these operations in sheet metal partitioned rooms, introducing properly tempered outside air to the rooms, and providing adequate means for exhausting the air without disturbing the conditioned air in the areas surrounding these operations.

## Maintenance

Maintenance and operation of the air conditioning equipment is carefully supervised on a 24 hr basis.

Typical penthouse control panel

Temperatures and humidity conditions are carefully noted from readings taken every two hours by attendants. An operating engineer is in attendance on the centrifugal compressors on each 8 hr shift, in addition to the operator assigned to handle temperature recording and control adjustments on the air handling units. It is estimated that approximately 900 man-hours per month are needed to maintain and operate the equipment; this figure includes the cleaning of 1200 capillary filters, lubrication, cleaning of equipment, minor repairs, the maintenance of all exhaust systems and ventilation fans in the building and a complete inspection of all penthouses and equipment three times daily.

## Engineering Building

The plant No. 2 engineering building, a two story, brick faced, cinder block structure, featuring glass block and very few windows, also is conditioned by a centrifugal type "Freon" compressor. Two 150 gpm centrifugal pumps circulate 45 F chilled water to two units housed in one standard penthouse, handling a total of 52,800 cfm. The heat load of this installation is approximately 75 tons and 67,000 sq ft of floor space is conditioned.

A constant air condition is maintained to keep large scale drawings and tracings at a standard size, as well as for human comfort. Ceiling type outlets were used for air distribution in conjunction with a 9 ft 0 in. ceiling, a condition that made it necessary to carry a slightly higher discharge temperature than would otherwise be necessary to avoid draft difficulties. The two units are identical in general design to units used on the main installation, but use steam coils for heating.

On both installations, gage and temperature readings are taken every two hours and then recorded in logs provided for the purpose. Accurate logs of total hours operated, cost per hour, day, month and year, and unit efficiencies, are maintained for reference and as aids to better future operation.

## Operating Costs

In passing, it is interesting to note that, disregarding amortization of initial investment, depreciation, and the pumping costs of delivering condenser water to the compressors, the average operating costs of the air conditioning system at plant No. 3 on the basis of a 30 day operating month are approximately as follows:

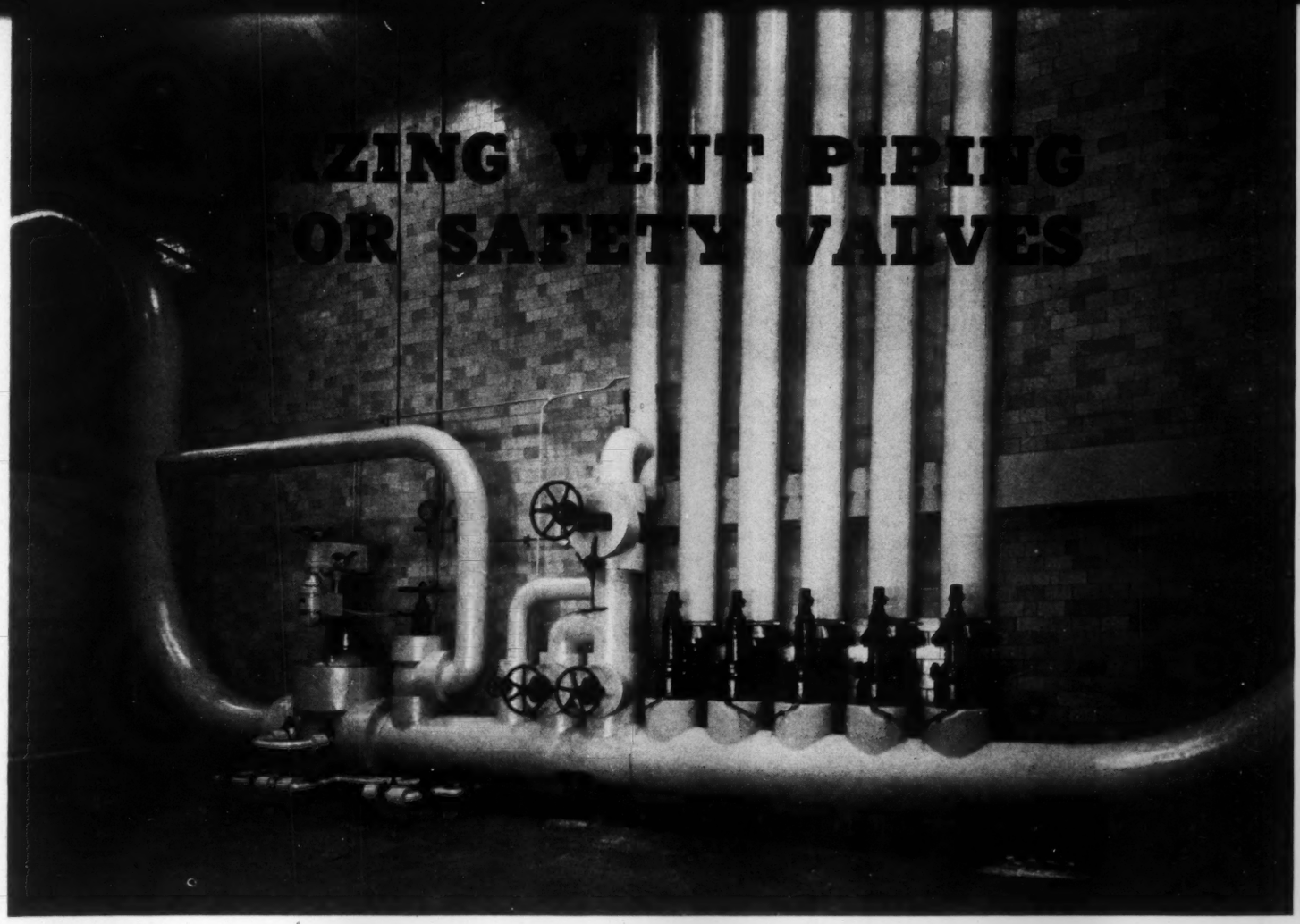
| | |
|---|---|
| Cost of operating fans | $ 3,323 |
| Cost of operating circulating pumps | 756 |
| Cost of operating compressors | 7,956 |
| Cost of maintenance and operation (labor) | 540 |
| Total cost per month | $12,575 |

These figures represent a cost of approximately $17.46 per manufacturing hour, and with 3000 employees per shift, bring the cost per employee per hour to $0.0058. From the employee relation standpoint as well as the precision manufacturing standpoint, the management feels that the air conditioning installation at the Allison Div. is definitely aiding in maintaining closer manufacturing tolerances and better employee efficiency . . . and hence is essential to the manufacture of the Allison aircraft engine.

## CALIBRATE RANGE FINDERS AT 75 F, 50 PER CENT HUMIDITY

Air conditioning is making it possible to test and calibrate long range artillery and anti-aircraft range finders, telescopes and other types of optical instruments under actual combat conditions in a specially constructed calibration room at the government's Raritan, N. J. arsenal.

Constant temperature and humidity conditions necessary to accurately calibrate delicate mechanisms are maintained in the calibration room by a 10½ ton air conditioning system.




Safety valve and vent pipe installation on the exhaust from a topping turbine

# Max W. Benjamin, Engineer, The Detroit Edison Co., Presents Better Method Than the "Rules of Thumb"

PIPING designers who have faced the problem of sizing vent lines for steam safety valves are well aware that for many years the only basis of design available to them has been the ASME boiler code requirement that "when a discharge pipe is used, the cross-sectional area shall be not less than the full area of the valve outlet or of the total of the areas of the valve outlets discharging thereinto, and shall be as short and straight as possible and so arranged as to avoid undue stresses on the valve or valves," or alternatively a rule of thumb method which calls for a vent line one pipe size larger than the safety valve. In many cases this system results in a satisfactory solution. There are, however, other situations in which these rules are of doubtful value, and vent lines could be designed with more confidence if a rational basis were employed. Examples of such cases are those in which the vent piping is exceptionally long or where several valves discharge into a common vent duct. Other circumstances probably will come to the minds of designers who have dealt with the problem.

**SIZING VENT PIPING for steam safety valves is a problem which often may be solved successfully by the ASME boiler code requirement or by rules of thumb. There are, however, many situations where these rules are of doubtful value, and where such piping could be designed with more confidence if a rational basis were employed. Such a method is given here. . . . The method described is in part a practical application of the acoustic velocity limitations in the flow of elastic fluids through conduits of uniform cross-sectional area. In addition to applying the acoustic velocity principle to the flow inside the vent pipe, the author introduces the idea that where an umbrella fitted vent is used it is necessary that the velocity pressure in the steam jet leaving the safety valve be at least equal to the static pressure inside the vent pipe. . . . Not only have vent lines designed by this method proved satisfactory in operation, but tests on safety valve capacity have yielded incidental data on the vent piping which show that the method is sound**

In addition to the unusual features of some vent line installations, the high ratio of relieving capacity to size which is provided by some of the newer safety valve designs, or the inherently high capacity resulting from high set pressures, may make it desirable to disregard the above rules and design the vent piping on the basis of a more detailed analysis.

This article presents a method for calculating the size of safety valve vent piping which it is hoped will be useful in those special cases which require careful consideration. It has been used with success on several unusual cases in the author's experi-

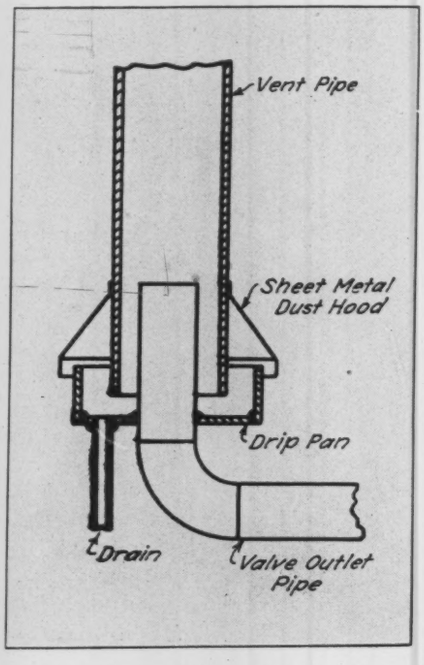
ence during the last six or eight years and is regularly employed as a check on the thumb rule even in rather ordinary cases.

## Design Requirements

There are two important factors which must be kept in mind in the design of safety valve vents. First, the vent must have sufficient capacity to carry away all the steam passed by the valve. Second, the vent line must be so constructed that the thermal expansion, which accompanies the passage of steam, will not throw stresses into the valve or its connection to the protected vessel. While it does not concern the vent design it may be mentioned parenthetically that the flow of steam through the safety valve itself may cause stresses and distortions of serious magnitude. This results from the high kinetic energy of the steam jet leaving the valve and its reaction in making the turn through the valve outlet elbow. For this reason the outlet elbow should be connected as close to the valve as possible, thus reducing the moment arm at which the reactive forces occur.

Designs to accommodate thermal expansion are of two general types, each of which presents a problem with reference to vent line capacity. The first design for flexibility involves some sort of flexible but continuous connection between the valve outlet and vent pipe. Slip joints and spiral flexible tubing have been used for this purpose. The other design for flexibility, which has been used quite generally, is the discontinuous "umbrella" fitting, such as illustrated in Fig. 1.

Fig. 1—A typical umbrella fitting



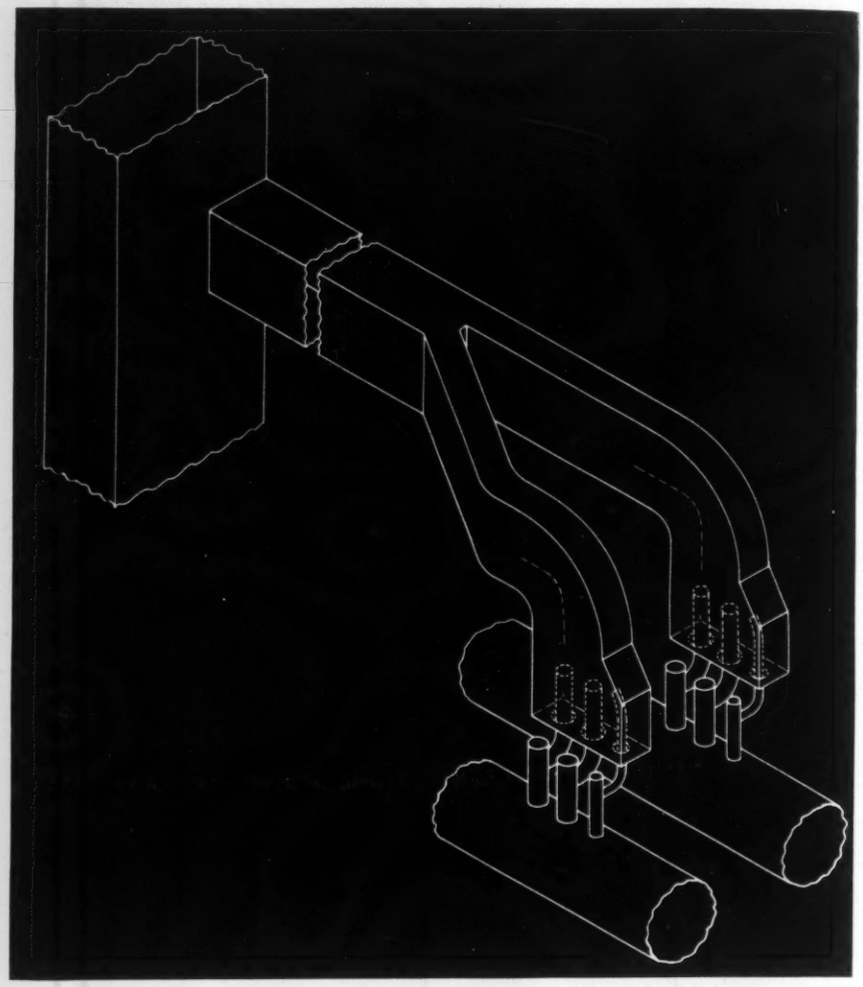



Fig. 2—Safety valves and vent ducts on a 550,000 lb per hr, double barrel, pressure reducing station

Use of a slip joint connection insures against steam blowing into the room which houses the safety valve, but this type of joint may stick and thus fail to protect the valve against undue strains. From the capacity standpoint the problem is merely one of calculating the pressure which will exist at the valve outlet so as to be sure the vent pipe construction is adequate and that the valve capacity will not be impaired because of high discharge pressure.

The umbrella type of fitting with its generous clearance between valve and vent pipe insures freedom from sticking and consequent thermal expansion strains. On the other hand, if the vent line is not designed properly, steam will blow out into the room, constituting at least an annoyance or possibly a hazard either to adjacent equipment or to personnel.

## Acoustic Velocity

The author's article, How to Design Steam Piping for Maximum Capacity and High Pressure Drop, which appeared in HPAC for September, 1936, explained in some detail the phenomenon of acoustic velocity in the flow of an elastic fluid through conduits of uniform diameter. It was pointed out there that the maximum velocity which the fluid can attain in such a conduit is equal to the velocity of sound in the fluid. This limiting velocity will occur at the outlet end of the pipe, since the specific volume of the fluid will be greatest at the outlet end as a result of pressure drop in the direction of flow.

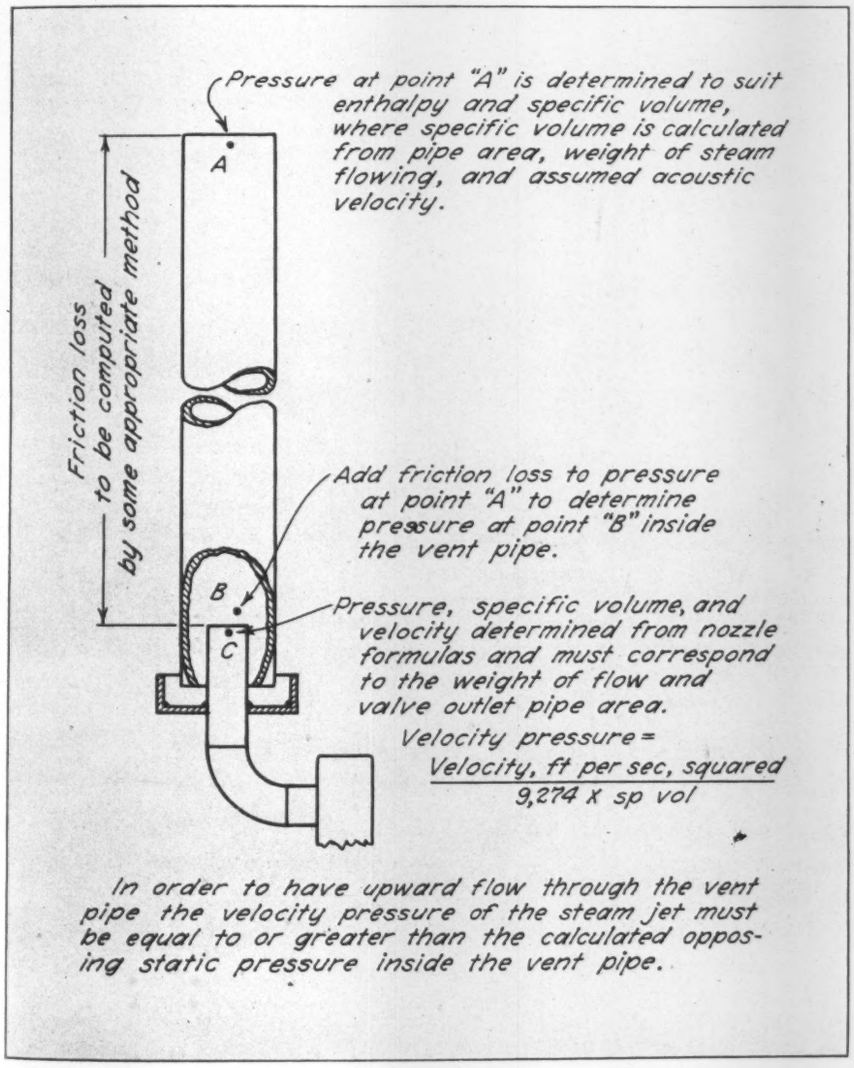
In safety valve vents of the sizes ordinarily used, it frequently happens that when the safety valve is relieving at its maximum capacity the acoustic velocity actually is attained. The important point here is that in order to accommodate the weight of steam flowing and not exceed the acoustic velocity (which is physically impossible) *the pressure at the outlet end of the vent will automatically adjust itself to produce a specific volume which in conjunction with the weight of flow and cross-sectional area of the vent pipe will produce the acoustic velocity.* If the vent pipe is of relatively small diameter, the specific volume of the steam will have to be relatively small and consequently the pressure at the outlet end may be considerably above atmospheric pressure. Pressure inside the entrance to the vent pipe will be equal to the outlet pressure plus the friction loss from entrance to outlet. The final drop in pressure to that of the surrounding atmosphere will take place after the steam is discharged from the pipe.

## The Vent Problem

The vent line problem already has been partially indicated. Irrespective of the means employed to obtain flexibility the pressure inside the inlet end of the vent should be determined so that the thrust on the pipe can be calculated. This may be particularly important if the vent pipe contains offsets. Also, if a slip joint or flexible tube connection is used between the valve and its vent, it is well to know the pressure against which the valve will blow. With high set pressures, the discharge pressure will have little influence on the capacity of the valve, but for lower set pressure, say up to 100 psi gage, the use of a long, small diameter vent may result in valve discharge pressures which will reduce the valve capacity. While no general rule can be laid down that will apply with equal accuracy to all makes and styles of safety valves, it is believed that in no case should the calculated discharge pressure imposed on the valve by the vent piping be allowed to exceed 40 per cent of the set pressure. With a well designed steam nozzle the capacity is not affected until the discharge pressure is about 58 per cent of the initial pressure. Some safety valves are designed like nozzles up to the seat and disc, but from seat to valve outlet the approximation to a nozzle is very poor. In addition to the possible effect on capacity it is important to recognize that in installations where several valves discharge into the same duct, as illustrated in Fig. 2, a high back pressure in the vent may produce undesirable disturbances in the pressures at which successive valves will pop.

For a vent line equipped with an umbrella fitting, calculation of the static pressure at the inlet to the vent is only half of the problem. The other half is to determine whether steam will blow out of the fitting and into the room. In answer to this question the author's experience supports the theory that if the velocity head of the steam jet leaving the safety valve is equal to or greater than the calculated static pressure inside the vent inlet the design is satisfactory. If the velocity head leaving the valve is less than the calculated static pressure inside the vent inlet it is quite likely that steam will blow out through the umbrella fitting. The velocity head leaving the valve is beyond the piping designer's control, but he can reduce the pressure inside the vent by increasing the diameter of the piping. If the vent pipe is larger than necessary, air will be aspirated into the vent and will be blown out with the steam—which is not objectionable if the vent pipe is not wastefully large.

**Fig. 3—Factors to be considered in umbrella fitted vent pipe size calculation**


Pressure at point "A" is determined to suit enthalpy and specific volume, where specific volume is calculated from pipe area, weight of steam flowing, and assumed acoustic velocity.
A
Friction loss to be computed by some appropriate method
Add friction loss to pressure at point "A" to determine pressure at point "B" inside the vent pipe.
B
Pressure, specific volume, and velocity determined from nozzle formulas and must correspond to the weight of flow and valve outlet pipe area.
C
Velocity pressure = Velocity, ft per sec, squared / 9,274 X sp vol
In order to have upward flow through the vent pipe the velocity pressure of the steam jet must be equal to or greater than the calculated opposing static pressure inside the vent pipe.

## Solution of Problem

A certain amount of trial and error is required in the calculation of a vent size by this method and

it is believed that the solution of a sample problem is the best way to illustrate what needs to be done. Let the assumed problem be as follows:—To design an umbrella fitted vent line for a safety valve on a superheated steam main carrying steam at 900 F. The valve is set to relieve at 850 psi gage and the rated capacity of the valve is 44,000 lb per hr. Diameter of the valve outlet is 5 in. The vent line has an equivalent length of 120 ft, including unavoidable turns.

*Solution*—Assume pressures of 50, 25, 10 and 5 psi gage inside the valve discharge elbow and determine the size of valve outlet needed for each of these pressures as indicated in the table below. These calculations assume the valve has an overall nozzle efficiency of 30 per cent in the conversion of available heat drop into velocity. The assumption of an overall nozzle efficiency is necessary because test data of this sort are not available; it would be interesting to have factual efficiency data either from valve manufacturers or from tests conducted in users' plants. The effect of assuming the efficiency too high will be to select a vent size that is too small, and while the assumed value of 30 per cent may be low, it has resulted in adequate vent design in cases with which the author is familiar.

| Initial Pressure Psi g | Pressure inside discharge elbow Psi g | Intial Enthalpy Btu per lb | Adiabatic End Point Enthalpy Btu per lb (1) | Available Energy Drop Btu per lb (2) | Actual Energy Drop Btu per lb (3) |
|---|---|---|---|---|---|
| 850 | 50 | 1453 | 1174 | 279 | 83.8 |
| 850 | 25 | 1453 | 1137 | 316 | 94.8 |
| 850 | 10 | 1453 | 1102 | 351 | 105.2 |
| 850 | 5 | 1453 | 1087 | 366 | 109.8 |

| Pressure inside discharge elbow Psi g | Actual End Point Enthalpy Btu per lb (4) | Velocity ft per sec (5) | Approx Sp Vol Cu ft per lb (6) | Reqd dia of valve outlet, inches (7) $d = 0.226\sqrt{\frac{\text{lb/hr} \times \text{sp vol}}{\text{ft/sec}}} = 47.4\sqrt{\frac{\text{vol}}{\text{vel}}}$ |
|---|---|---|---|---|
| 50 | 1369 | 2050 | 10.3 | 3.36 |
| 25 | 1358 | 2180 | 16.45 | 4.11 |
| 10 | 1348 | 2300 | 25.8 | 5.03 |
| 5 | 1343 | 2340 | 32.0 | 5.55 |

1) Adiabatic end point read from Mollier diagram. Assumes isentropic drop from initial conditions to assumed pressure inside the discharge elbow.

2) Available energy drop = initial enthalpy minus adiabatic end point enthalpy.

3) Actual energy drop = available energy drop × assumed overall nozzle efficiency.

4) Actual end point enthalpy = initial enthalpy minus actual energy drop.

5) Velocity, ft per sec = $223.7\sqrt{\text{actual energy drop}}$

6) Specific volume, cu ft per lb, is interpolated from steam tables for assumed discharge pressure and calculated actual end point enthalpy.

7) Required valve outlet diameter as derived by equating total volume per unit time to the product of outlet area in square feet times velocity in feet per unit time, i.e.

$$\text{volume} = \text{area} \times \text{velocity}$$

$$\text{lb per hour} \times \text{sp vol} = \frac{\pi d^2}{4 \times 144} \times \text{velocity, ft per sec} \times 3600$$

$$d^2 = \frac{576}{3600\,\pi} \times \frac{\text{lb per hr} \times \text{sp vol}}{\text{velocity, ft per sec}}$$

$$d = 0.226\sqrt{\frac{\text{lb per hr} \times \text{sp vol}}{\text{velocity, ft per sec}}}$$

From this calculation it appears that the pressure at the 5 in. diameter outlet of the valve will be approximately 10 psi gage, the velocity of steam issuing from the valve will be 2300 ft per sec and its specific volume will be about 25.8 cu ft per lb. Velocity head in pounds per square inch for any fluid is equal to $v^2d/9274$, where $v$ = velocity in ft per sec, and $d$ = density in lb per cu ft = 1/sp vol. Therefore the velocity head in the steam leaving the valve = (2300 × 2300) ÷ (9274 × 25.8) = 22.1 psi. Hence, for an umbrella fitted vent to suit this particular application the static pressure inside the inlet to the vent including friction loss and static pressure at the vent outlet must not exceed 22.1 psi gage.

The vent line pressure is calculated as follows. Assume that the steam velocity at the vent outlet is the acoustic velocity. As a first trial value the acoustic velocity may be taken as 100,000 ft per min, although its true value varies with pressure and specific volume according to the equation

$$\text{acoustic velocity} = 60\sqrt{gnPV}$$

*where* $g$ is gravitational acceleration in ft per sec per sec; $n$ is the adiabatic exponent which has a value of about 1.13 for saturated steam and 1.3 for superheated steam; $P$ is the final absolute pressure in pounds per square foot, and $V$ is specific volume at pressure $P$. Values of acoustic velocity which are sufficiently accurate for the purpose of this type of problem may be read from Fig. 4.

Assume that a 6 in. diameter vent line were fitted to this valve. With the pipe size, weight of flow, and velocity all known, there is only one condition of specific volume that can exist inside the outlet end of the vent; thus specific volume is equal to

$$\frac{\text{pipe area, sq ft} \times \text{acoustic velocity, fpm}}{\text{flow in lb per min}}$$

With the assumed 6 in. vent, specific volume is equal to

$$\frac{\frac{6.065^2 \times \pi}{4 \times 144} \times 100{,}000}{\frac{44{,}000}{60}} = 27.4 \text{ cu ft per lb}$$

With an enthalpy of about 1340 Btu per lb (the approximate value at the safety valve outlet minus a small allowance for change in velocity as the steam flows through the vent pipe) the pressure corresponding to a specific volume of 27.4 cu ft per lb is interpolated from the steam tables as about 23.3 psi absolute or 8.6 psi gage.

From this point it is necessary to calculate the pressure rise back to the vent inlet. In all such cases, where the velocity of flow is very high, the overall length of the pipe should be divided into several short sections for the purpose of calculating the friction loss. The reason for this is that the relatively high pressure change corresponding to the high velocity produces variations in specific volume which should be taken into account. Otherwise the calculated pressure rise will be too high if calculated from the outlet back to the inlet, or too low if calculated in the direction of flow. For this problem let the vent line be divided into four sections of 30 ft each. Using Fritzsche's formula the friction loss calculation is as follows:

$$p = \frac{0.8LF^{1.85} \times \text{sp vol}}{d^{4.97}}$$

*where* $p$ = friction loss, psi; $L$ = equivalent length of pipe, ft; $F$ = steam flow, lb per sec; and $d$ = inside diameter of pipe, in. Then for a vent pipe of 6.065 in. inside diameter, $p =$

$$\frac{0.8 \times \left(\frac{44{,}000}{3600}\right)^{1.85} \times \text{sp vol} \times 30}{6.065^{4.97}} = 0.317 \text{ sp vol.}$$

in which the specific volume to be used is the average value in the corresponding section of the line. By making several trials the appropriate average specific volume for the section at the outlet end is found to be 24.

*Then* $p = 0.317 \times 24 = 7.6$

$p_2 = p_1 + p = 23.3 + 7.6 = 30.9$ psi absolute

sp vol at $p_2 = 20.6$

$$\text{Average sp vol} = \frac{27.4 + 20.6}{2} = 24$$

(to check the average sp vol assumed for this section)

For the next section

$p = 0.317 \times 18.9 = 6.0$

$p_3 = p_2 + p = 30.9 + 6.0 = 36.9$ psi absolute

sp vol at $p_3 = 17.2$

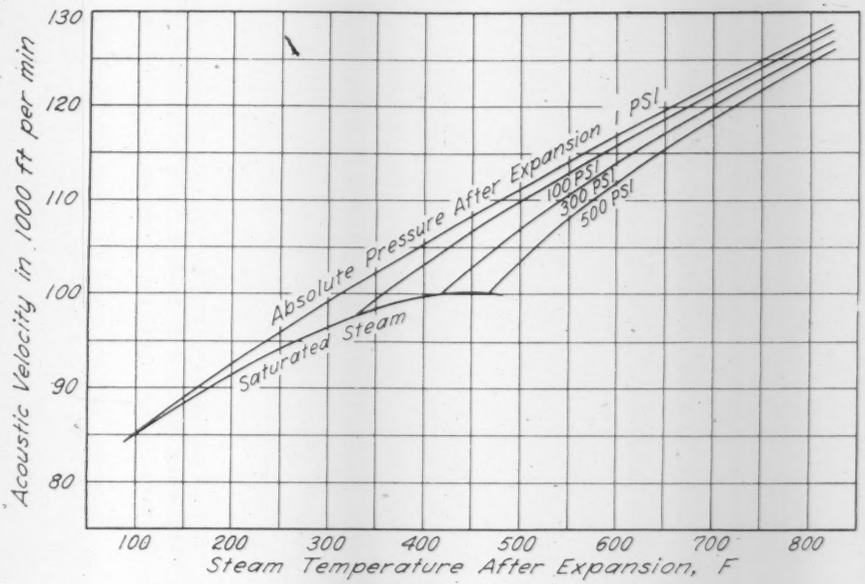



Fig. 4—Approximate values of acoustic velocity for steam

$$\text{Average sp vol} = \frac{20.6 + 17.2}{2} = 18.9$$

For the next section

$p = 0.317 \times 16.2 = 5.13$

$p_4 = p_3 + p = 36.9 + 5.13 = 42.03$ psi absolute

sp vol at $p_4 = 15.2$

$$\text{Average sp vol} = \frac{17.2 + 15.2}{2} = 16.2$$

For the section at the vent inlet,

$p = 0.317 \times 14.4 = 4.56$

$p_5 = p_4 + p = 42.03 + 4.56 = 46.59$ psi absolute

sp vol at $p_5 = 13.6$

$$\text{Average sp vol} = \frac{15.2 + 13.6}{2} = 14.4$$

Thus the pressure inside the vent inlet is calculated as 46.59, or say 46.6 psi absolute or 31.9 psi gage. Therefore, as computed for this example, the 6 in. vent line would be unsatisfactory for use with an umbrella fitting because the velocity head at the safety valve outlet was calculated at only 22.1 psi, which is not sufficient to overcome the vent line pressure of 31.9 psi gage just determined.

Repeating the pressure drop calculations using an 8 in. vent pipe produces the following results:

Pressure at vent outlet = 14.7 psi absolute. (In this case the pressure at the vent outlet corresponding to the acoustic velocity was found to be less than atmospheric. Therefore the actual velocity, with no air inspirated into the vent, will be lower than the acoustic and the vent outlet pressure actually will be atmospheric.)

Pressure at end of first section = 17.9 psi absolute.

Pressure at end of second section = 20.6 psi absolute.

Pressure at end of third section = 23.0 psi absolute.

Pressure inside vent inlet = 25.1 psi absolute = 10.4 psi gage.

With a velocity head of 22.1 psi in the steam at the safety valve outlet it is evident that all resistances imposed by the 8 in. vent pipe will be overcome and an umbrella fitting can be used without fear of steam blowing out into the room.

It is acknowledged that this method still requires the exercise of a certain amount of judgment, especially in the matter of the overall nozzle efficiency assumed for the safety valve. The need for making this assumption will be eliminated only when safety valve test data in terms of overall nozzle efficiency are made available. Beyond that, however, the method does account for variables which do not appear in the older rules of thumb and it may therefore be considered a better approach to the problem.

A number of successful installations have been made using this system, some of which required larger size vents than would have been called for without the more detailed study.

**New catalogs, bulletins, etc. are reviewed in the department "Trade Literature" which is published in HPAC monthly**

# "Open for Discussion"

## Communications to the Editor

# WHY PANEL HEATING?

## F. W. Hutchinson Presents a Part of the Case for Panel Heating . . . Opinion *Pro* and *Con* Requested

PARADOXICALLY, the rapid acceptance of low temperature panel heating in Europe was probably largely due to a characteristic of the system which, in terms of effective heating, is inconsequential; the English and continental home owner and architect are strongly averse to any visible form of heating surface and were therefore ready and willing to adopt a form of heating which provides adequate warmth while remaining completely invisible. Thus, the impetus responsible for the prompt acceptance of low temperature radiant heating abroad apparently was derived from architectural rather than engineering advantage.

However, as the number of panel installations increased and operating data became available, a number of specific engineering characteristics of panels came to the foreground and increased attention was given to radiant heating by engineers and heating specialists. Most significant of these observations was that with low temperature heating panels in the ceiling a condition of substantial comfort can be realized in a room with air temperature lower than that needed when radiators or warm air grilles are the sources of heat. The extent to which a reduction in air temperature is possible depends upon the construction of the building and the outside temperature, but European experience has shown that a reduction of 5 to 8 F is not unusual.

### Humidity Indoors

Aside from the obvious economic advantages of reduced inside air temperature — lower heat loss due to infiltration or to ventilation and, in some cases, reduction in transmission losses—there seem to be certain intangible advantages in terms of comfort and health. Among these are said to be higher indoor relative humidity, without use of a humidifier, and a feeling of increased "freshness" of the air.

The effect of lower temperature on relative humidity can be easily visualized by referring to the simplified psychrometric chart reproduced here. If saturated outside air at 40 F (point *a*) were heated to 72 F (point *b*) the resultant relative humidity would be 32 per cent, while if heating ceased at 62 F the relative humidity would be 45 per cent (point *c*). Then, for an outside condition as assumed, each degree reduction of the inside air temperature is responsible for an increase of approximately 1.3 per cent in the indoor relative humidity. The magnitude of this term is clearly a function of the absolute humidity only and the effectiveness of lower room air temperature as a means of increasing relative humidity therefore varies directly as the outside temperature and the outside relative humidity.

In cold weather the improvement due to inside temperature reduction is slight (0.1 per cent per degree for —10 F, 100 per cent relative humidity outside) while at moderate outside temperature the effectiveness is pronounced (1.9 per cent per degree at 50 F, 100 per cent).

The above characteristic is in part advantageous, but it also represents a distinct disadvantage. Since moderate temperatures exist for a relatively large fraction of the heating season the value of decreased drying of the inside air is realized over the longer interval; but when the outside temperature becomes very low—the time when "drying" of the inside air is normally most severe—the value of panel heating as a means of raising the relative humidity is practically negligible.

**WHY PANEL HEATING?, a question asked in a letter published in the September HPAC, is discussed here. The intent of the author, who is assistant professor of mechanical engineering at the University of California, is to review some of the intangible factors which influence the selection of a given type of heating system and to show from psychrometrics, and by reference to current ideas regarding "freshness" of air in an enclosure, the background for some of the claims which have been advanced for panel heating. . . . Possible improvement in the indoor relative humidity (without use of a humidifier) is shown to be negligible during very cold weather, but quite pronounced for moderate outside conditions. . . . Cooler room air reduces the "shock" of leaving an inside warm environment since the danger of anemia of the mucous membranes of the nasal passage is reduced; however, no quantitative significance can yet be assigned to this effect. . . . Since the heating surfaces in a panel warming installation are at a very much lower temperature than are the surfaces used with other types of heating systems, the "baking" effect on the air is reduced, and there is reason to believe that this factor may be significant in improving the degree of "freshness" which can be realized in a closed space. . . . Mr. Hutchinson thus summarizes some of the arguments for panel heating. The editors will welcome other comment both *pro* and *con***

### Air Temperature

Effects on health or comfort of inside air temperature cannot be stated with exactitude. Within the limits of present knowledge it is possible only to call attention to some observed effects and to conditions which are believed to be of significance.

More important than the actual

inside air temperature is the difference between inside and outside temperatures and the resultant "shock" experienced by the occupant as he leaves a heated room. In a warm room the membranes of the nose are seemingly flooded with blood and lymph and covered with a thick secretion; when sudden chilling occurs the blood vessels of the nose constrict, reducing the blood supply and lowering the temperature of the mucous membrane which remains swollen with lymph. Some authorities believe that this anemic condition of the enlarged membrane is conducive to infection. With radiant heating the internal membranes of the nose are subject only to energy transfer from the cooler room air and the anemia resulting from sudden exposure to low outside temperature should therefore be reduced. The quantitative significance of this factor cannot now be evaluated.

## Ion Content of Air

It is generally recognized that there is some vitalizing property of outdoor air which has not yet been duplicated indoors. Many observers believe that this property may be associated with the ion content of the air. Ions are positively or negatively charged molecule clusters and are roughly classified as large or small. In nature they are produced by cosmic rays, solar radiation and by radioactive elements in the soil; their production artificially can be accomplished by electrical means and equipment for this purpose is available commercially.

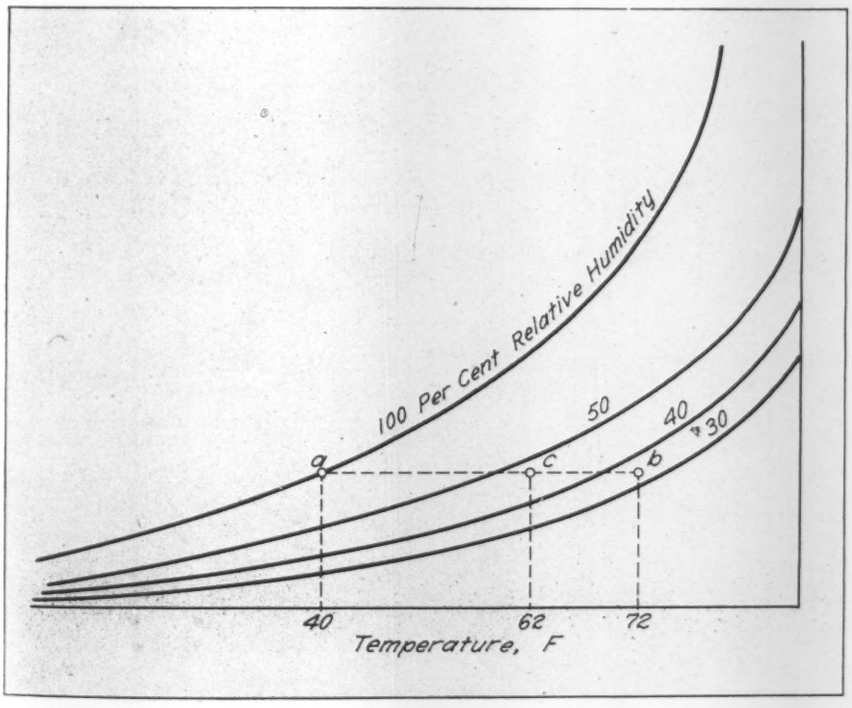
Present ideas concerning the effectiveness of ions in producing either "freshness" or "stuffiness" are vague, but two vital facts are evident: (1) The passage of outside air through ventilating ducts and through heating or air conditioning equipment is responsible for material changes in the ion content of the treated space. (2) A preponderance of small positive ions is known to have an adverse effect on the ability of the occupants to think clearly and is responsible for headaches and nose irritations. Small negative ions, on the other hand, predispose to relaxation, increase mental activity and have other desirable characteristics.

The whole subject of ionization is in such a vague state that it is not now possible to say what value, if any, may eventually accrue from deliberate control of ion concentration by artificial means. The importance of the subject to present day heating and air conditioning derives from the fact that equipment now in use is itself a factor in the determination of the ionic state of the conditioned space.

Radiant heating is designed to establish comfort with a minimum of air heating and such heating as does occur results from contact of the air with a surface at much lower temperature (90 to 100 F) than is customary with other systems. Therefore panels might be expected to have less of a "baking" effect on the air in the room and effect to a minimum the ion content.

European experience seems to demonstrate that a greater feeling of "freshness" is experienced in a panel heated structure than in one heated by straight convection. Limited experience in this country leads to the same result. Whether this is a result of the ionic state, or, more simply, due to lower air temperature is not known. In this connection it is interesting to note that Dr. Bedford, in summarizing the results of a study on subjective impressions of freshness, lists seven requirements for a pleasant and invigorating atmosphere; the first of these requirements is that the room be as cool as is compatible with comfort.

Simplified psych chart




## "Open for Discussion"

**Readers of HEATING, PIPING AND AIR CONDITIONING are invited to contribute their views to these "Open for Discussion" pages, which will appear from time to time. Comments on articles we have published, expressions of timely interest on developments in design, installation, operation or maintenance of heating, piping or air conditioning systems, other remarks of value to HPAC's readers — all will be welcomed by . . . . THE EDITOR.**

# Why Draft Gage Readings on Boiler Furnaces Sometimes Seem Incorrect


## B. H. Jennings, Professor of Mechanical Engineering at Northwestern Technological Institute, Explains a Puzzling Paradox, Tells How to Make Corrections

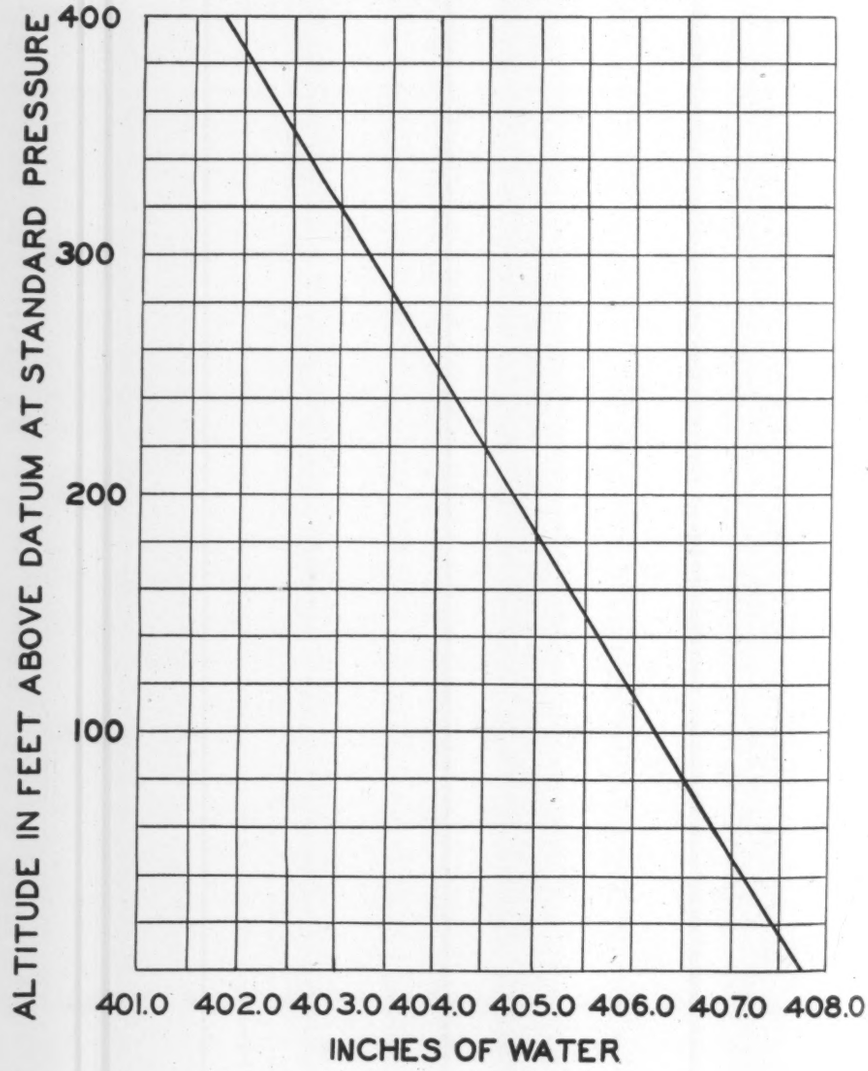

IN analyzing test results from boiler furnaces and gas heat transfer equipment, data frequently are found which indicate values for pressure loss in the gas passages that seem erroneous or even implausible. An example appeared recently in some test results on an oil burning boiler: The draft on leaving the boiler at the entry to the stack was 0.08 in. of water *negative* and the draft in the combustion chamber before entering the boiler passes was 0.10 in. of water *negative*, according to the data. This apparently indicated that the combustion gases were moving from a point of lower absolute pressure in the combustion chamber to a point of higher pressure at the stack inlet, [—0.08 — (—0.10)] or + 0.02 in. of water. A pressure increase in such a case is certainly contrary to expectation. The draft gages on the test had previously been carefully checked (and even interchanged) with no variation in the readings. Later, during a test at different capacity, the same unit was put under a *positive* pressure of 0.06 in. of water in the combustion chamber and the leaving draft gage indicated 0.075 in. of water *positive* pressure —confirming the puzzling paradox of the previous test by a pressure increase, in this case, of 0.015 in. of water.

**DRAFT GAGE readings sometimes yield data that seem erroneous or even implausible. Prof. Jennings explains here that this is due to the difference in the mean temperature of the combustion gases in the furnace and the temperature of the outside atmospheric air—provided the measurement taps for the draft readings are placed at different elevations. . . . A correction chart and explanation of its use are presented on pp. 625-626**

The explanation of this paradox is not difficult. However, so little has been written on this phase of drafts that a discussion of the topic and presentation of a set of correction curves seem worthwhile. The situation is not unique to oil burners but holds true for coal stokers and grates as well. For the solid fuels the draft loss across the grate is usually so large that the paradox remains hidden, but false overall pressure drops from ashpit to breeching must occur unless suitable corrections are applied to the difference in draft gage readings.

It should be noted that in accordance with conventional practice, draft has been taken to mean the pressure difference existing between the air outside and the gases inside the furnace at any particular point. With natural draft, pressures inside the furnace are less than those outside so a draft gage indicates a pres-

Fig. 1—Altitude-pressure graph for 59 F air at standard pressure (14.696 psi = 407.74 in. of 70 F water)



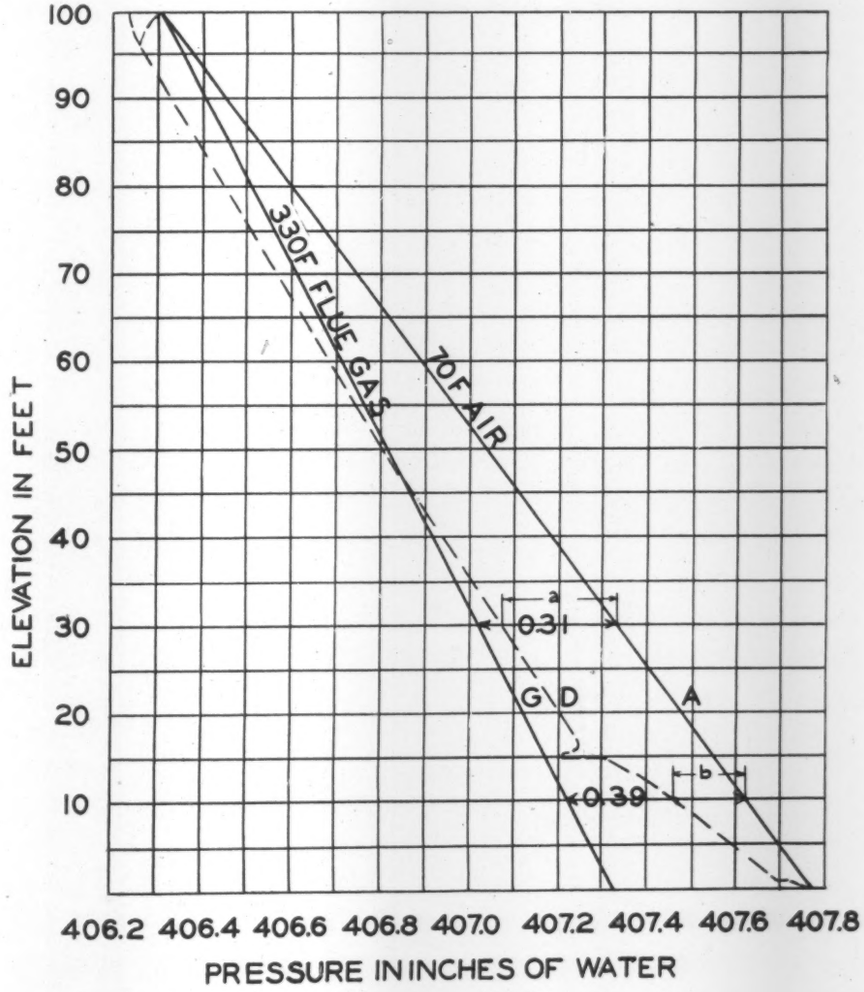
sure less than atmospheric, really negative in character.

## Cause of Errors

Incorrect pressure difference readings between two draft gages can occur whenever the mean temperature of the combustion gases in the furnace is different from that of the outside atmospheric air provided, of course, the measurement taps for the draft readings are placed at different elevations. The pressure of the atmosphere varies directly with elevation, as is well known; this is shown in Fig. 1. The pressure in the furnace and stack also varies directly with the elevation in addition to any influences of frictional pressure loss and kinetic effects. But if the temperature and thereby the density of the furnace gas is different from that of the outside atmospheric air it is obvious that even with no flow in the furnace and consequently no frictional pressure drop there exists for two gages attached at different elevations a differential pressure indication between their readings.

Consider the pressure gradient with altitude which would exist in a chimney filled with hot gases if the flow were momentarily stopped, as by the closing of a tight damper. Fig. 2 (curve *G*) shows the pressure gradient through 100 ft for such a static condition in a chimney and boiler filled with flue gas at an assumed average temperature of 330 F. Curve *A* shows the variation in pressure with altitude for the air outside the furnace, in this case taken for a 70 F temperature. Using these curves for illustrative purposes, draft gage readings between conditions inside and outside the chimney are taken at, say, 10 ft and 30 ft. At 10 ft the draft is 0.39 in. (negative) and at 30 ft the draft is 0.31 in. (negative). Commonly the difference in draft gage readings between two points is taken to indicate the pressure loss because of frictional effects of flow. In this case a frictional flow loss of [—0.31 — (—0.39)] or +0.08 in. of water is indicated, but as there is no flow obviously the +0.08 in. is not a frictional flow reading—certainly not a gain as here indicated.

Fig. 2—Pressure-elevation graph for 70 F air and 330 F flue gas. (The flue gas density is taken as 4 per cent greater than air)




Suppose flow is taking place in a chimney with gases at 330 F; the shape of the pressure curve is not like curve *G* in Fig. 2 but more like the dash line *D*. Even with flow there is still the column of hot gas in the chimney and as our reference is always to the atmosphere as a datum, how much of the measured draft at 30 ft (*a* in.) and at 10 ft (*b* in.) is used in overcoming frictional resistances to flow? Obviously, it is not merely the (—0.26 in.) at *a* less the (—0.16 in.) at *b* or —0.10 in. difference, because the +0.08 in. density factor is still present even if hidden. The true frictional loss corrected for elevation effects is really [—0.10 — (+0.08)] = —0.18 in. The minus 0.18 in. indicates a pressure drop of that absolute value. The 0.08 is here taken as minus because the density effect in question automatically makes the pressure readings indicate an increase so the sign must be made minus to make the correction.

The curve *D* really represents the loci of draft readings, the horizontal distance between curves *A* and *D* at any elevation being the draft gage reading in each case. Because of kinetic energy effects, *D* may have unusual inflections as when gases are passing through restricted passageways (dampers, baffles) at high velocity. In such cases the pressure becomes less in absolute value at the points of high velocity moving sharply to the left and partly recovering pressure as velocity decreases after passing the obstacle. Such a condition is indicated on *D* at 15 ft elevation. Near the outlet from the stack, also, the gases moving at some velocity pass from the stack and approach zero velocity above the stack outlet with a slight pressure increase. Because of this velocity effect curve *D* can cross over *G* as shown in Fig. 2.

Fig. 3 shows a boiler furnace with draft gages in place. As has been

shown, the reading of gage $X$ minus the reading of gage $Y$ for the pressure loss through the furnace passes, is not correct unless a correction is made for the effect of the varying hot-gas column inside and the cold-air column outside. Nor will gage $Z$ give a correct value for pressure loss unless the same correction is made. In fact, gage $Z$ connected as shown will read exactly the same value as that of $X$ minus $Y$.

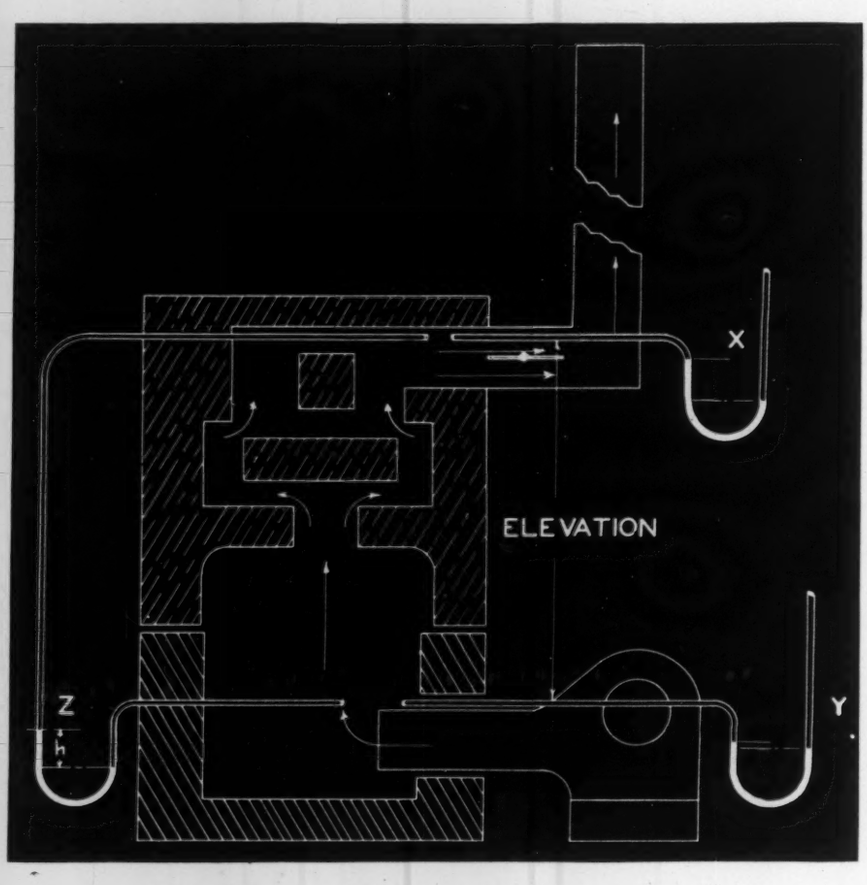



Fig. 3—Diagrammatic furnace with draft gages attached

## The Proof

To prove these statements, first consider gage $Z$. If there were no flow and the temperatures of gas inside and outside the furnace were the same, there would be no deflection and $h$ would be zero. But with no flow and hot gas inside the furnace a deflection would occur because the heavier column of cooled and therefore denser gas in the tubing outside the furnace exerts a greater pressure than the hot and less dense gas inside the furnace. (This would be opposite to the direction shown during flow and only takes the position shown when the frictional resistance is adequately high to offset this temperature elevation effect.) Gage $X$ reads the same value no matter where it is located vertically, provided the gas in the furnace closely resembles air in its density characteristics. The gas in the left or attachment leg of gage $X$ outside the furnace takes the same temperature as that of the atmosphere and consequently if the gage $X$ were lowered to an elevation such as that of $Y$ there would be an increase in atmospheric pressure with lowered elevation on the right leg but this would be exactly balanced out by the gas column in the tubing cooled to atmospheric temperature in the left leg. Thus gage $X$ reads the same whether high or low on the boiler for the same tubing inlet point. Furthermore when $X$ is placed at the level of $Y$ as the atmospheric pressure is the same for both $X$ and $Y$ it is obvious that reading $X$ minus reading $Y$ must equal the reading $h$ of gage $Z$. It is perhaps unnecessary to note that variations in horizontal displacements of gage tubing or of gas channels through a furnace can have no effect on the gage readings caused by gas density variations.

Correction curves have been plotted in the graph on p. 626 to show the real magnitude of pressure changes when these are made independent of temperature and elevation effects. In use their application is simple, as explained on p. 625.

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**A new type of shatterproof, non-combustible side wall and roof construction combining three types of glass fiber and prefabricated steel panels is being used to insure insulation and acoustical control for working efficiency in the twin 4000 ft long army bomber assembly plants being erected at Fort Worth, Tex., and Tulsa, Okla. Engineers of the Austin Co., who designed both plants, evolved the ingenious combination in an entirely new approach to the task of insulating vast factory areas for economical year 'round air conditioning.**

**With a total of 406 carloads of fiber glass required for these jobs—203 carloads for each plant—they will probably contain more glass in their windowless steel side walls and roofs than the largest daylight factories ever built. By blanketing all interiors with several layers of these materials, which are calculated to eliminate or control all condensation and to reduce heat transference to a practical minimum, the cooling load has been substantially reduced. The plants are so large, however, that each requires 7000 tons of refrigeration for air conditioning.**

**The photo shows workmen unrolling some of the white glass wool which blankets roofs and walls. Runner channels, which are welded to the steel purlins 33 in. on centers, support the 3/8 in. ribbed expanded metal lath on which is spread a thin, light reflecting mat of fiber glass and a 3 in. blanket of white fiber glass, which affords sound absorption, fire resistance and insulation. This, in turn, is covered by a metal deck, some of which is already in place in the foreground. Sections of this rigid metal deck serve as temporary working platform for the men who are placing the fiber glass mat and insulation in this view. Note the manner in which they are lapping the edges to insure complete coverage.**

**Several layers of a special vapor seal paper, asphalt and fiber glass insulating board are still to be applied above this metal deck and the entire surface will be covered by waterproof materials.**

# Correction Chart for Draft Readings

**By B. H. Jennings**
**Professor of Mechanical Engineering**
**Northwestern Technological Institute**

AS explained in some detail in my article on p. 622 of this month's HPAC, incorrect pressure readings between two draft gages used in running tests on boiler furnaces and gas heat transfer equipment can occur whenever the mean temperature of the combustion gases in the furnace is different from that of the outside air (provided, of course, the measurement taps for the draft readings are placed at different elevations).

The chart on the next page gives correction curves which may be used to find the real magnitude of the pressure changes independent of temperature and elevation effects. The use of these curves is quite simple:

1) The draft gage reading at the downstream point in the gas flow is used and recorded with its appropriate sign, *plus* if greater than atmospheric and *minus* if less than atmospheric.

2) The draft gage reading at the upstream (earlier) point of gas flow is also recorded with its appropriate sign.

3) The difference when (2) with its proper sign is algebraically subtracted from (1) represents the uncorrected pressure drop.

4) For the difference in elevation between (1) and (2) and for the average flue gas temperature read the correction from the chart on the next page. This is algebraically *minus* when (1) is at higher elevation than (2).

5) Algebraically combine (3) with (4). A *minus* sign indicates a pressure drop which is to be expected in the direction of flow.

6) In the less usual case when the downstream point (1) is lower in elevation instead of higher than (2) the correction in (4) from the chart algebraically carries a *plus* sign.

An example will illustrate the method of applying the correction. The flue gases leaving the third pass of a boiler show a pressure of —0.10 in. of water (*i.e.*, a pressure less than atmospheric) and the draft tap is 18 ft above boiler floor level. The pressure over the fire is —0.09 in. of water and the draft tap is 6 ft above boiler floor level. The average gas temperature is considered as 1400 F through the passes in question. It is desired to find the frictional pressure drop through the boiler passes.

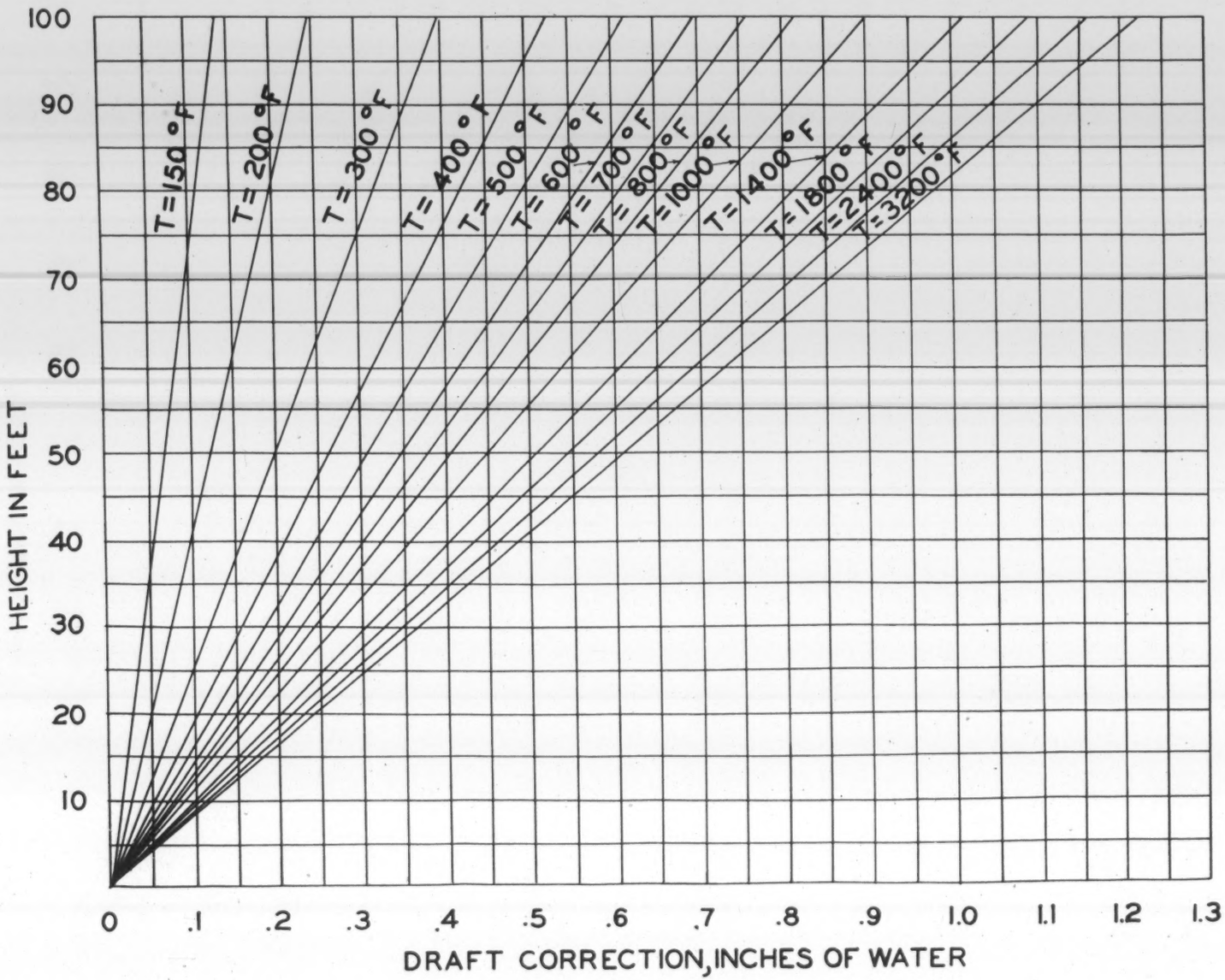
*Solution*

[—0.10 — (—0.09)] = —0.01 in. of water is the uncorrected pressure drop.

Reading from the chart for 1400 F flue gas shows a correction of —0.12 with 70 F air and at an elevation of 18 — 6 or 12 ft. The corrected pressure drop is thus (—0.01 —0.12) = —0.13 in. of water. The true pressure drop is therefore 0.13 in. of water in absolute magnitude.

This example indicates how extreme the inaccuracy can be when corrections are not made to the draft gage readings. This may be particularly important when comparisons of pressure drops through different portions of a boiler setting are considered. The total pressure drop indicated for two points in a boiler setting can be less than that indicated between intermediate points if different elevations of the tap points exist. Bad restrictions in pass design and baffling may be experimentally obscured and lead to false conclusions unless a proper interpretation of the data obtained is made.

[See explanation of use on preceding page]




Draft-loss correction chart, based on 70 F air and flue gas 4 per cent heavier than air. Correction is *minus* when downstream gage is at higher elevation than upstream gage. *T* is flue gas temperature

# INDUSTRY'S GREATEST TRANSPORTATION SYSTEM

## Improved Process Piping Aids Distilling at Calvert . . . . . By S. J. Paradiso

**INDUSTRY'S greatest transportation system is—without a doubt—piping. Judged by the number of miles of thoroughfare, amount and variety of materials conveyed and its importance to production, the piping installed in industrial plants represents a most vital highway for getting things from one place to another. . . . Each different type of plant requires many piping facilities more or less common to all plants, and in the majority of cases piping which is more or less specialized to meet its particular needs is also essential. But even these piping systems designed for specialized functions have many features which can often be adapted to advantage in other industries. . . . For this reason, the piping at the Louisville plant of the Calvert Distilling Co. for the manufacture of pure grain spirits and whiskies, which is described here by Mr. Paradiso of Calvert, will be of interest to all concerned with the design, installation, the operation and maintenance of "industry's greatest transportation system"**

IN developing the processes for manufacturing pure grain spirits and whiskies at the Louisville plant of the Calvert Distilling Co., it was necessary to install piping of many different materials and of unusual and special design.

Rigid control is exercised in handling the materials in process from the time they evolve as a grain mash until they are finally packaged as a distilled alcoholic beverage. Cleanliness, sanitation and optimum conditions are of utmost importance in processing the grains and in eliminating contamination caused by undesirable organisms.

Specially fabricated wide sweep fittings and connections were installed in order to eliminate traps and pockets in the piping. Specially fabricated plug and diaphragm controlled valves were used. These valves are of bronze so that the alcoholic liquors will not be discolored.

### Piping Identification

In designing the process piping system, Calvert engineers complied with government regulations which require all pipe lines carrying whiskey, alcohol, alcoholic vapors, mash, fermented mash, stillage, steam, water, fusel oil, compressed air and compressed carbon dioxide to be erected so that the lines may be traced easily. A further requirement is that all such pipe lines be painted certain designated colors according to the material which passes through each. These colors are the following:

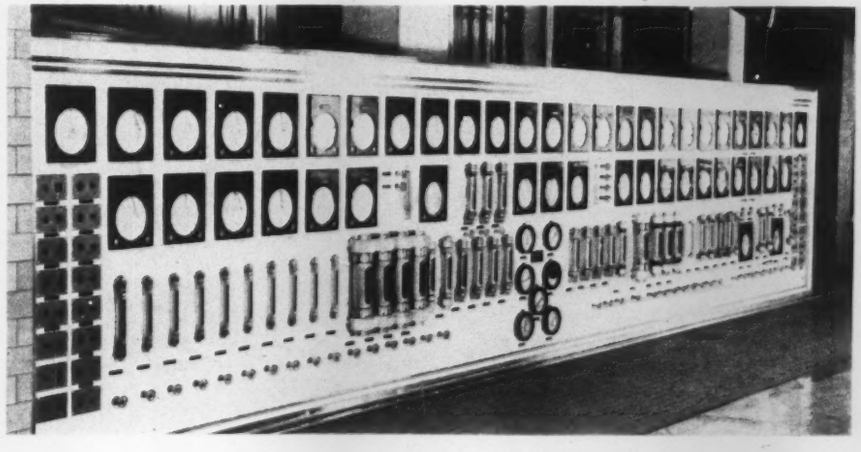
Red....Mash, beer or other distilling material.
Gray....Molasses or other fermenting material.
Black....Whiskey, alcohol, finished product.
Blue....Vapor, high wine, low wine or unfinished spirit.
Brown ....Stillage.
Yellow....Fusel oil.
White....Water.
Aluminum....Steam.
Orange....Air.
Green....Carbon dioxide.

Furthermore, all lines carrying spirits of any kind from the stills to the high wine room storage tanks must be so constructed that no material may be removed from the pipe lines. All fittings, valves, etc., are under government lock and seal, and such fittings may be opened only under government supervision.

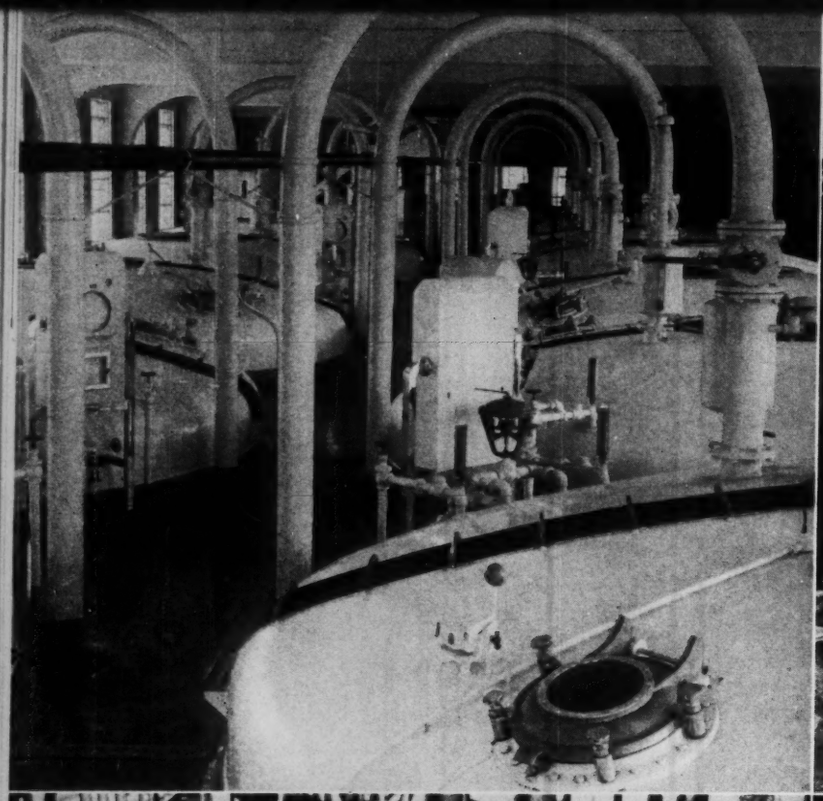
### Cleanliness Vital

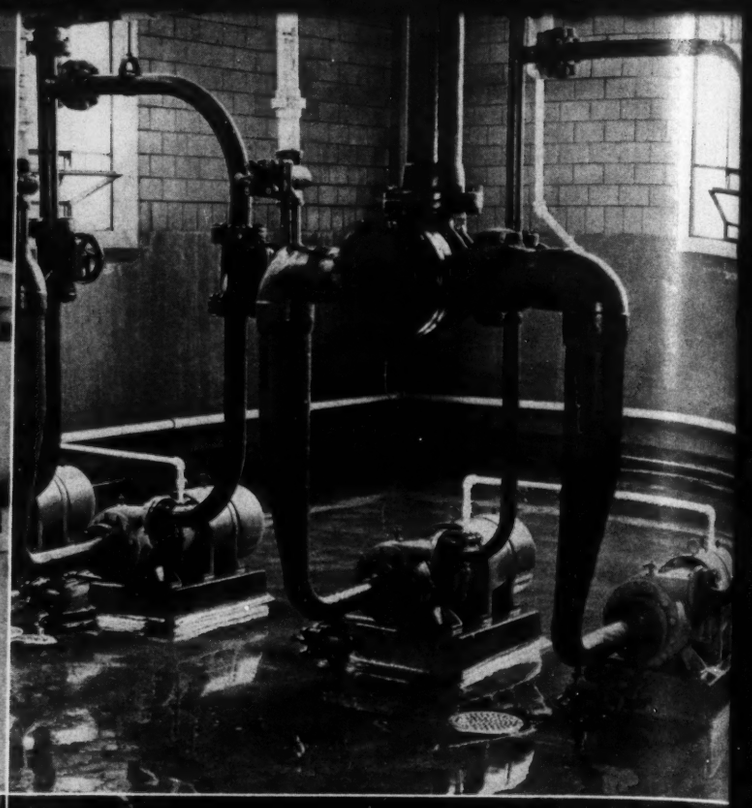
Copper piping is used in the fermenting room because of the acid characteristics of the mash. The alloy steel fermenting vessels are covered to prevent pollution by bacteria laden air. Since corners and crevices are ideal places for mash to lodge and breed undesirable bacteria, all of the piping leading to and from the fermenters is provided with distinctive wide sweep bends. The carbon dioxide draw-off lines coming from the tops of the fermenters are designed to allow a minimum pressure drop, and they are equipped with lubricated plug valves which prevent sticking.

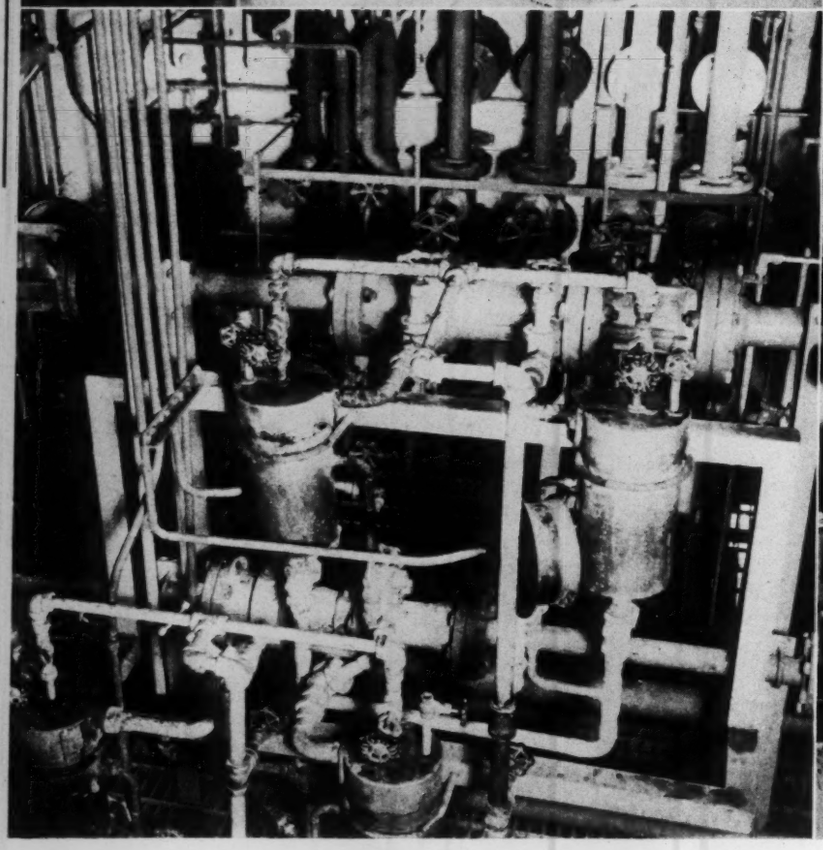
A dairy type pipe line is used to fill the fermenters; its special design

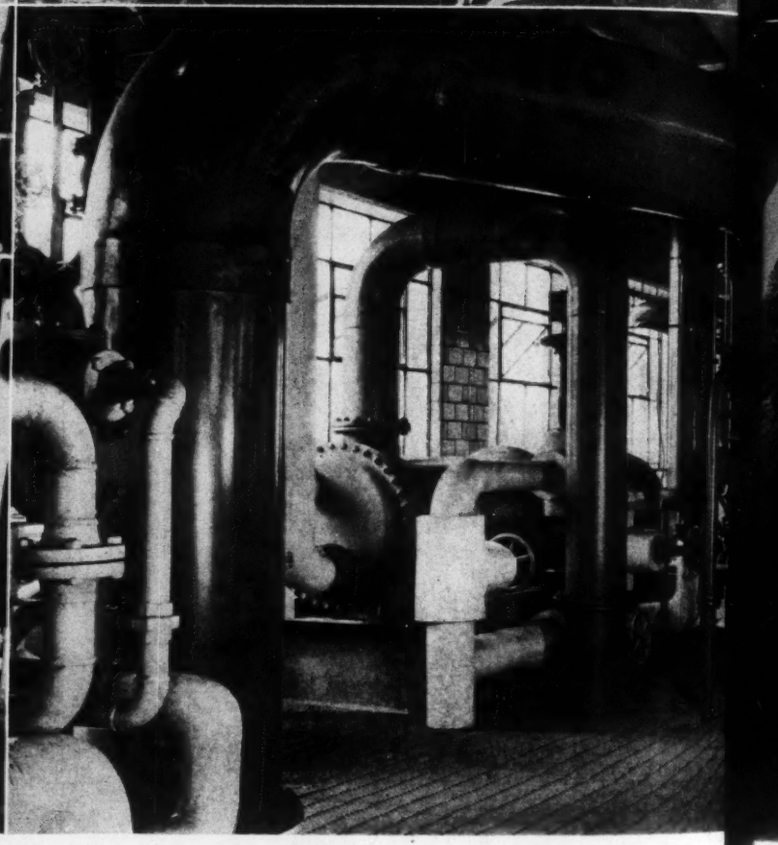


**The master control board. Many of the control lines are machined to three decimal figures in order to obtain accuracy**

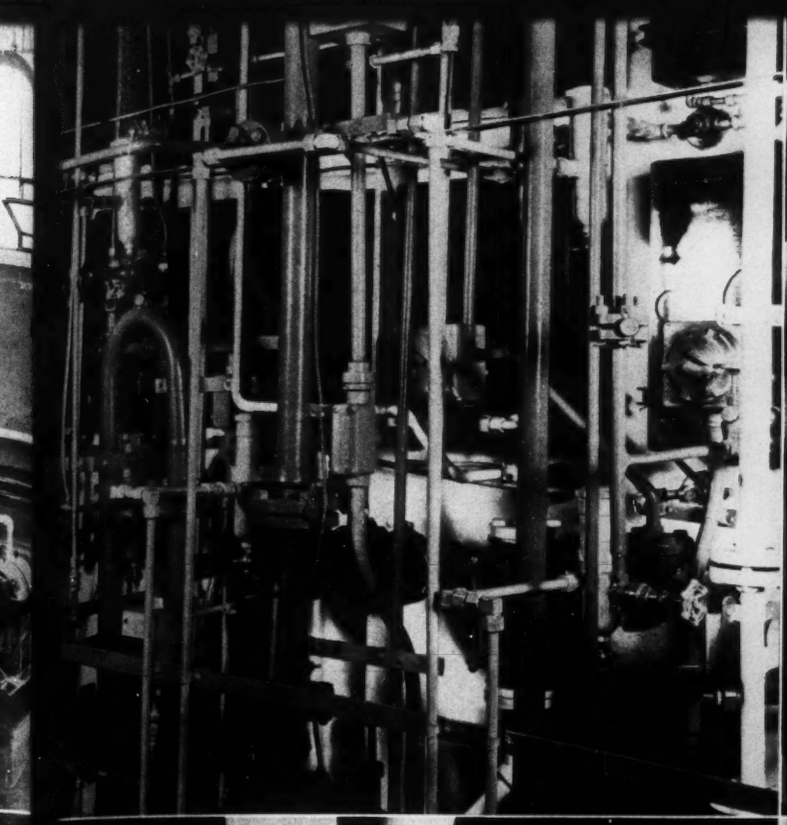


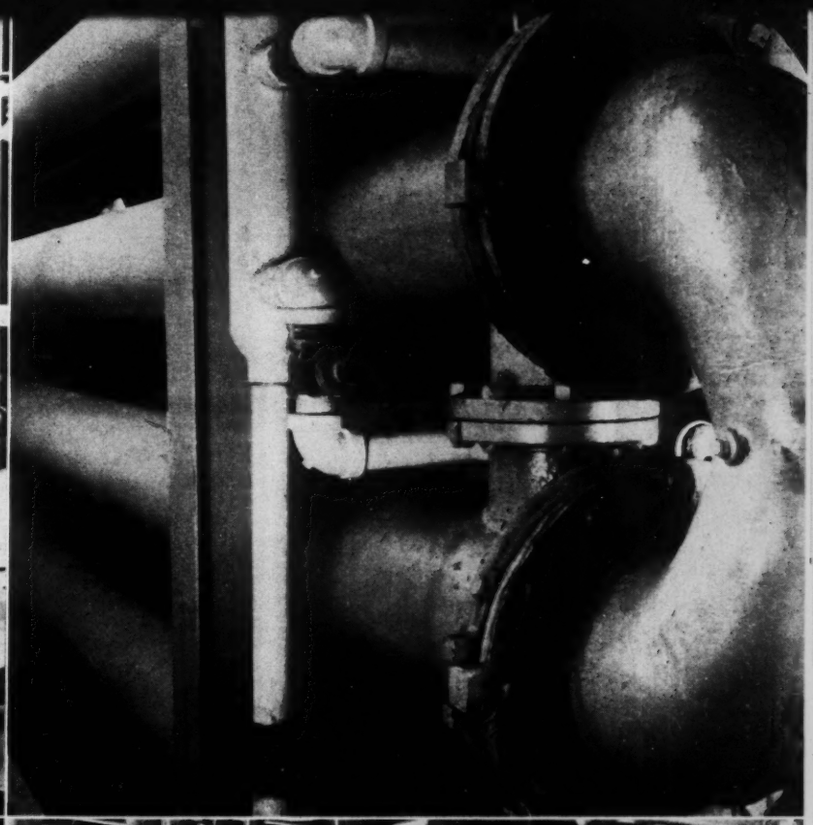


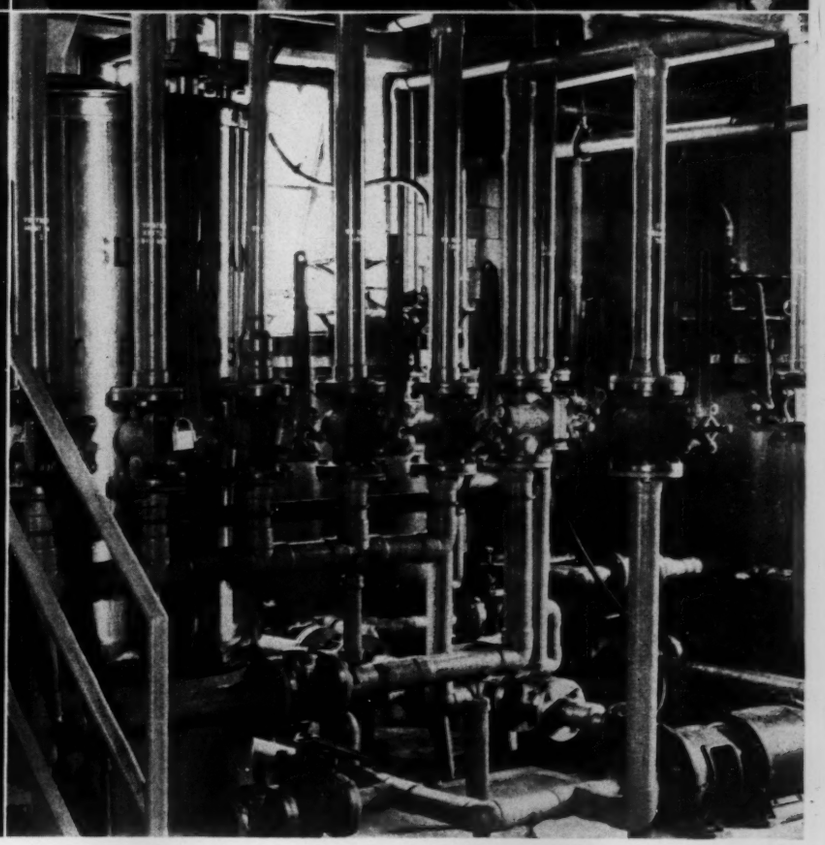




*Top left*—Fermenter room of the Louisville plant of the Calvert Distilling Co. Note the wide sweep bends on the carbon dioxide vapor draw-off lines and the lubricated plug valves. The dairy type fill line may be seen running down the length of the fermenter room between the banks of fermenters. . . . *Top right* — Feed pump installation. The gooseneck effect on the inlet side of the pumps prevents clogging. Note the diaphragm controlled and specially lubricated plug valves. . . . *Bottom left*—Venturi piping arrangements to measure the quantity of feed to the stills accurately. Control lines for the spirit unit are in the background. . . . *Bottom right*—Vapor draw-off lines from vacuum operated stills. The large line in the foreground is the draw-off line from the whiskey beer still. The smaller line in the background is the vapor draw-off line from the whiskey column which is operated at 68 mm Hg absolute pressure. The partially visible unit on the left of the photo is the double stage steam ejector which pulls the vacuum on the still units







*Top left*—Control lines from whiskey, gin and re-run stills to the master control board. . . . *Top right*—This exchanger is used to cool the cooked mash to the desired converting temperature. . . . *Bottom left*—View showing copper tubes of cooler. . . . *Bottom right*—A portion of the pump and filter room. Note the lubricated plug valves used on all lines and the special lock arrangement designed to comply with government regulations, as is described by Mr. Paradiso

makes it possible to transfer mash to any fermenter with the shortest possible length of pipe and without using a single valve.

A gooseneck fill spout is inserted in a convenient coupling connection in the dairy type line. The fill spout is placed in the side porthole of the fermenter, and the line is then ready for use. This arrangement makes it simple to keep the line clean, for there are a minimum number of bends to catch and collect mash. An additional requirement of the piping is that it must withstand the changes in temperature from the live steam at 12 psi pressure (used during sterilization) to the 65 to 72 F temperature of the cooled mash.

## Free Flow Essential

The elaborate automatic control system employed to operate the various still units would be useless if the lines leading to and from the units were not always free flowing. To insure this condition, Calvert engineers had to use novel equipment and dual pumps.

The difficult problem of maintaining the flow from feed tanks to stills was solved by using piping of a wide sweep type which decreases gradually and uniformly in size to fit the feed pump. Diaphragm controlled valves are used on the discharge lines, and special lubricated plug valves are used to direct the flow of mash to the pump unit in service.

Fermented mash is fed to the stills through a venturi in order to measure accurately the quantity of beer passing into the stills. The piping had to be constructed so that the venturi reading would be dependable. The slurry-like consistency of the fermented mash made the problem difficult, and the final installation followed test performances of many designs. It was determined that diaphragm controlled valves give excellent service for this type of material.

The stills are operated at 32 mm Hg absolute pressure, and this high vacuum requires that a minimum pressure drop be maintained between the steam vacuum ejector and the column. Oversized fabricated copper vapor draw-off lines were installed on the stills to insure this condition. The large size of the unit presented problems of fabrication and sealing which were not encountered in smaller ones. All other vacuum operated units were provided with similar lines.

All of the automatic controls for the spirits, whiskey, gin, and re-run stills are conveniently placed on a centrally situated master control board. Operation and control of quantity of flow, temperature, pressures, specific gravity and purity are practiced. Many of the control lines are machined to three decimal figures in order to obtain the accuracy which is essential. Connections must withstand extreme pressures and vacuums, and there must be no corrosion in the control lines if the instruments are to be relied upon.

Finished distilled liquor is handled in stainless steel tubing. The liquor is pumped from still surge tanks to wine room tanks and thence to the cistern and blending rooms. The lines are equipped with lubricated plug valves, which not only prevent pocketing and subsequent denaturation of alcoholic spirits, but which also provide an easily operated liquid tight cutoff in the line. Wrench operated valves, clearly marked with raised perforations on the plug to indicate which side is "blanked" off—and, consequently, which side is open—enable the operator to tell by inspection whether a valve is on or off.

In addition to transporting materials, tubing is used in heat exchangers. It was selected for its thermal conductivity and for its characteristic chemical reactions to the materials flowing in or around the tubing.

Advancements in process piping have played no small part in the development of the distilling industry. Extensive installations of modern process piping have been justified by the increased yields which have been attained. Higher standards of quality in regard to uniformity, odor and taste have been direct results of the newly developed process which depends upon the many specially fabricated pieces of piping and fittings.

## Quoted . . .

"Those who have been through a tropical hurricane comment on the extraordinary feeling of depression in the central lull and the exhilaration and excitement when the storm is at its height. When the air is lifeless, the heat oppressive, and the sky a coppery yellow, people find even routine exertions exhausting. Then when the hurricane reaches its crescendo of roaring winds, lashing rain, crashing timbers and tossing trees, there is a wild impulse to be up and doing. One feels strong as a giant.

"Just at a guess we should say that the cause was the difference in air pressures, resulting in a variation in oxygen intake similar to that experienced on high mountains. There may be a similar variation between hot and cold weather, although on a smaller scale. More oxygen per breath, quite as much as the need to keep warm, may account for the vigor of autumn.

"For all man's vaunted powers of reason, he is in considerable measure—and much more than most city folk realize—suspended in the natural balance of a world chemistry beyond his control. The turn of the seasons, the heat and cold, the rainfall and sunshine, the winds sweeping the thin envelope of air that coats the earth, are all part of a gigantic symphony. If modern science understands the playing of some of the instruments, we all of us none the less dance to the tune." —*Royal F. Munger, in the Chicago Daily News.*

* * *

A new method of air conditioning hospital operating rooms — which makes it possible to "bathe" surgeons, attendants and nurses in conditioned air during an operation, yet prevents "drafts" on the patient—has been installed in the St. Francis hospital, Poughkeepsie, N. Y. Outlets were arranged so that the air flows down the walls and across the room at the floor, rising in the middle of the room around the operating table. This makes it possible to keep surgeons and attendants comfortable and at top efficiency throughout the operation, yet does not allow direct air motion on the patient.—*Edward T. Murphy, vice-president in charge of marketing, Carrier Corp.*

# Storage Tanks for Cooling Systems Where Compressor Capacity Is Less Than Load

## William Goodman Presents Data to Simplify Selection of Compressor and Storage Tank

THERE are many places where cooling is required for only a few hours each day. In such places, it is sometimes possible to make worthwhile savings in air conditioning operating costs by using a storage tank. This is particularly true if the electrical demand charge is a substantial portion of the operating cost. In this case, the storage tank would be used together with a compressor whose capacity is less than the cooling load—the compressor operating for a period of time much longer than the duration of the cooling load. Thus, cooling might be required for only 4 hr of the day but the compressor might be allowed to operate perhaps 20 hr a day, the excess cooling capacity being stored in the tank.

### Sizing System

Table 1 is of value in computing the size of compressor and storage tank required for storage type systems. This table shows the ratio of the required compressor capacity to the cooling load, and the storage capacity required for various hours of compressor operation. The temperature of the water stored in the tank changes and consequently the compressor capacity also changes as the water is cooled. Therefore, when using this table, the *average* compressor capacity should be used. Note particularly when using the table that the storage capacity is given in Btu per ton of load—not in Btu per ton of compressor capacity as in the table on p. 503 of the August HPAC.

The following example illustrates the use of Table 1:

**Example 1**—If sufficient refrigerating capacity were provided for the maximum cooling load, a compressor of 75 tons would be required. Inasmuch as cooling is required for only 6 hr per day, a storage tank and a small compressor are to be installed. The compressor is to be allowed to operate 22 hr per day. The highest allowable temperature of the water in the storage tank is to be 48 F and the water is to be allowed to cool to 40 F.

Assume that when the water temperature is 48 F, the suction temperature of the compressor will be 42 F; and when the water temperature is 40 F, the suction temperature will be 34 F.

Find: (a) The refrigerating capacity required at the average suction temperature of 38 F. (b) The size of storage tank required.

*Solution*—

a) Referring to Table 1 in Column 12, for a 22 hr compressor operating time, find

$$\frac{\text{Compressor Capacity}}{\text{Load}} = 0.273$$

As the cooling load is 75 tons, the compressor capacity = 0.273 × 75 = 20.5 tons.

This compressor capacity of 20.5 tons is the average compressor capacity required at the average suction temperature. The highest suction temperature will be 42 F and the lowest suction temperature 34 F. Consequently, the average suction temperature is 38 F. The compressor capacity of 20.5 tons should be selected at the average suction temperature of 38 F.

b) From Column 13 of Table 1, for a 22 hr compressor operating time, the storage required is 52,370 Btu per ton of load. Referring to the table on p. 505 of the August HPAC, for an 8 deg rise in water temperature and for a storage of 52,300 Btu, the water storage required is 785 gal per ton of cooling load. Therefore, the total storage required = 785 × 75 = 58,875 gal.

When small compressors are to be run for long hours for cooling loads of relatively short duration, large chilled water storage tanks will be required. To eliminate the need for such large tanks, ice storage tanks are frequently used. The ice is frozen on coils submerged in the tanks. For the same storage capacity, the size of an ice storage tank will be considerably smaller than a chilled water tank. On the other hand, the suction temperatures required to freeze ice are lower than those required for chilling water. This article describes only the use of chilled water tanks.

Table 1 covers compressor operating times up to 24 hr. In some installations, a compressor may be operated for only a few hours in perhaps two or three days. For such installations, the compressor capacity and the storage capacity required may be computed by means of the following equations:

$$\frac{H_C}{H_L} = \frac{\theta_L}{\theta_C}$$

$$H_S = 12{,}000\,\theta_L \left(1 - \frac{\theta_L}{\theta_C}\right)$$

Note that in the above equations, the compressor running time and the cooling load period are in *hours*, whereas they are in minutes for the equations in the article on pp. 502-505 in the August HPAC.

**STORAGE**—**There are many places where cooling is required for only a few hours each day. In such cases, it is sometimes possible to make worthwhile savings in air conditioning operating costs by using a chilled water or ice storage tank. This is particularly true if the electrical demand charge is a substantial portion of the operating cost. . . . Bill Goodman, consulting engineer, The Trane Co., and a member of HPAC's board of consulting and contributing editors, presents here data to simplify computation of the compressor size and the storage tank required. Examples show in detail the use of the method**

### Symbols

$c$ = specific heat of stored liquid.
$H_C$ = refrigerating capacity of the compressor, tons
$H_L$ = cooling load, tons
$H_S$ = thermal energy stored in tank, Btu.
$s$ = specific gravity of stored liquid.
$t_1$ = initial temperature of liquid in storage tank, F.
$t_2$ = final temperature of liquid in storage tank, F.
$\theta_C$ = running time of compressor, hr.
$\theta_L$ = duration of cooling load in conditioned spaces, hr.

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## Table 1—Compressor and storage capacities
(Storage capacities are in Btu per ton of *load*)

| Compressor Operating Time | Duration of Cooling Load — Hours | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 1 | | 2 | | 3 | | 4 | | 5 | | 6 | | 7 | | 8 | |
| Hrs. | Comp. Cap. Load | Storage Capacity | Comp. Cap. Load | Storage Capacity | Comp. Cap. Load | Storage Capacity | Comp. Cap. Load | Storage Capacity | Comp. Cap. Load | Storage Capacity | Comp. Cap. Load | Storage Capacity | Comp. Cap. Load | Storage Capacity | Comp. Cap. Load | Storage Capacity |
| 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | 10 | 11 | 12 | 13 | 14 | 15 | 16 | 17 |
| 1 | 1.000 | 0 | | | | | | | | | | | | | | |
| 2 | 0.500 | 6,000 | 1.000 | 0 | | | | | | | | | | | | |
| 3 | 0.333 | 8,000 | 0.667 | 7,999 | 1.000 | 0 | | | | | | | | | | |
| 4 | 0.250 | 9,000 | 0.500 | 12,000 | 0.750 | 9,000 | 1.000 | 0 | | | | | | | | |
| 5 | 0.200 | 9,600 | 0.400 | 14,400 | 0.600 | 14,400 | 0.800 | 9,600 | 1.000 | 0 | | | | | | |
| 6 | .167 | 10,000 | .333 | 16,000 | .500 | 18,000 | .667 | 16,000 | 0.833 | 10,000 | 1.000 | 0 | | | | |
| 7 | .143 | 10,280 | .286 | 17,140 | .429 | 20,570 | .571 | 20,570 | 0.714 | 17,140 | 0.857 | 10,290 | 1.000 | 0 | | |
| 8 | .125 | 10,500 | .250 | 18,000 | .375 | 22,500 | .500 | 24,000 | 0.625 | 22,500 | 0.750 | 18,000 | 0.875 | 10,500 | 1.000 | 0 |
| 9 | .111 | 10,670 | .222 | 18,670 | .333 | 24,000 | .444 | 26,670 | 0.556 | 26,660 | 0.667 | 24,000 | 0.778 | 18,660 | 0.889 | 10,670 |
| 10 | 0.1000 | 10,800 | 0.200 | 19,200 | 0.300 | 25,200 | 0.400 | 28,800 | 0.500 | 30,000 | 0.600 | 28,800 | 0.700 | 25,200 | 0.800 | 21,600 |
| 11 | .0909 | 10,910 | .182 | 19,640 | .271 | 26,180 | .364 | 30,550 | .455 | 32,730 | .546 | 32,720 | .636 | 30,540 | .727 | 27,270 |
| 12 | .0833 | 11,000 | .167 | 20,000 | .250 | 27,000 | .333 | 32,000 | .417 | 35,000 | .500 | 36,000 | .583 | 35,000 | .667 | 32,000 |
| 13 | .0769 | 11,080 | .154 | 20,310 | .231 | 27,690 | .308 | 33,230 | .385 | 36,920 | .462 | 38,770 | .539 | 38,770 | .615 | 36,920 |
| 14 | .0714 | 11,140 | .143 | 20,570 | .214 | 28,280 | .286 | 34,290 | .357 | 38,570 | .429 | 41,140 | .500 | 42,000 | .571 | 41,140 |
| 15 | 0.0667 | 11,200 | 0.133 | 20,800 | 0.200 | 28,800 | 0.267 | 35,200 | 0.333 | 40,000 | 0.400 | 43,200 | 0.467 | 44,800 | 0.533 | 44,800 |
| 16 | .0625 | 11,250 | .125 | 21,000 | .188 | 29,250 | .250 | 36,000 | .313 | 41,250 | .375 | 45,000 | .438 | 47,250 | .500 | 48,000 |
| 17 | .0588 | 11,290 | .118 | 21,180 | .177 | 29,650 | .235 | 36,710 | .294 | 42,350 | .353 | 46,590 | .412 | 49,410 | .471 | 50,590 |
| 18 | .0556 | 11,330 | .111 | 21,330 | .167 | 30,000 | .222 | 37,330 | .278 | 43,330 | .333 | 48,000 | .389 | 51,330 | .444 | 52,670 |
| 19 | .0526 | 11,370 | .105 | 21,470 | .158 | 30,320 | .211 | 37,900 | .263 | 44,210 | .316 | 49,260 | .368 | 53,050 | .421 | 54,740 |
| 20 | 0.0500 | 11,400 | 0.1000 | 21,600 | 0.150 | 30,600 | 0.200 | 38,400 | 0.250 | 45,000 | 0.300 | 50,400 | 0.350 | 54,600 | 0.400 | 56,400 |
| 21 | .0476 | 11,430 | .0952 | 21,720 | .143 | 30,860 | .191 | 38,860 | .238 | 45,710 | .286 | 51,430 | .333 | 56,000 | .381 | 57,860 |
| 22 | .0455 | 11,450 | .0909 | 21,820 | .136 | 31,090 | .182 | 39,270 | .227 | 46,360 | .273 | 52,370 | .318 | 57,270 | .364 | 59,090 |
| 23 | .0435 | 11,480 | .0870 | 21,910 | .130 | 31,310 | .174 | 39,650 | .217 | 46,960 | .261 | 53,220 | .304 | 58,440 | .348 | 60,260 |
| 24 | .0417 | 11,500 | .0833 | 22,000 | .125 | 31,500 | .167 | 40,000 | .208 | 47,500 | .250 | 54,000 | .292 | 59,500 | .333 | 61,330 |

| Compressor Operating Time | Duration of Cooling Load --- Hours | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 9 | | 10 | | 11 | | 12 | | 13 | | 14 | | 15 | | 16 | |
| Hrs. | Comp. Cap. Load | Storage Capacity | Comp. Cap. Load | Storage Capacity | Comp. Cap. Load | Storage Capacity | Comp. Cap. Load | Storage Capacity | Comp. Cap. Load | Storage Capacity | Comp. Cap. Load | Storage Capacity | Comp. Cap. Load | Storage Capacity | Comp. Cap. Load | Storage Capacity |
| 1 | 18 | 19 | 20 | 21 | 22 | 23 | 24 | 25 | 26 | 27 | 28 | 29 | 30 | 31 | 32 | 33 |
| 9 | 1.000 | 0 | | | | | | | | | | | | | | |
| 10 | 0.900 | 10,800 | 1.000 | 0 | | | | | | | | | | | | |
| 11 | .818 | 19,630 | 0.909 | 10,910 | 1.000 | 0 | | | | | | | | | | |
| 12 | .750 | 27,000 | 0.833 | 20,000 | 0.917 | 11,000 | 1.000 | 0 | | | | | | | | |
| 13 | .692 | 33,230 | 0.769 | 27,700 | 0.846 | 20,300 | 0.923 | 11,070 | 1.000 | 0 | | | | | | |
| 14 | .643 | 38,570 | 0.714 | 34,280 | 0.786 | 28,290 | 0.857 | 20,580 | 0.929 | 11,140 | 1.000 | 0 | | | | |
| 15 | 0.600 | 43,200 | 0.667 | 40,000 | 0.733 | 35,200 | 0.800 | 28,800 | 0.867 | 20,800 | 0.933 | 11,210 | 1.000 | 0 | | |
| 16 | .563 | 47,250 | .625 | 45,000 | .688 | 41,250 | .750 | 36,000 | .813 | 29,250 | .875 | 21,000 | 0.938 | 11,250 | 1.000 | 0 |
| 17 | .529 | 50,820 | .588 | 49,420 | .647 | 46,580 | .706 | 42,350 | .765 | 36,710 | .824 | 29,650 | 0.882 | 21,170 | 0.941 | 11,290 |
| 18 | .500 | 54,000 | .556 | 53,330 | .611 | 51,340 | .667 | 48,000 | .722 | 43,340 | .778 | 37,330 | 0.833 | 30,010 | 0.889 | 21,330 |
| 19 | .474 | 56,840 | .526 | 56,840 | .579 | 55,590 | .632 | 53,050 | .684 | 49,260 | .737 | 44,220 | 0.790 | 37,890 | 0.842 | 30,320 |
| 20 | 0.450 | 59,400 | 0.500 | 60,000 | 0.550 | 59,400 | 0.600 | 57,600 | 0.650 | 54,600 | 0.700 | 50,400 | 0.750 | 45,000 | 0.800 | 38,400 |
| 21 | .429 | 61,710 | .476 | 62,860 | .524 | 62,860 | .571 | 61,720 | .619 | 59,440 | .667 | 56,990 | .714 | 51,430 | .762 | 45,710 |
| 22 | .409 | 63,820 | .455 | 65,460 | .500 | 66,000 | .546 | 65,450 | .591 | 63,820 | .636 | 61,080 | .682 | 57,280 | .727 | 52,730 |
| 23 | .391 | 65,740 | .435 | 67,820 | .478 | 68,860 | .522 | 68,880 | .565 | 67,830 | .609 | 65,740 | .652 | 62,600 | .696 | 58,870 |
| 24 | .375 | 67,500 | .417 | 70,000 | .458 | 71,500 | .500 | 72,000 | .542 | 71,500 | .583 | 70,010 | .625 | 67,500 | .667 | 63,990 |

| Compressor Operating Time | Duration of Cooling Load --- Hours | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 17 | | 18 | | 19 | | 20 | | 21 | | 22 | | 23 | |
| Hrs. | Comp. Cap. Load | Storage Capacity | Comp. Cap. Load | Storage Capacity | Comp. Cap. Load | Storage Capacity | Comp. Cap. Load | Storage Capacity | Comp. Cap. Load | Storage Capacity | Comp. Cap. Load | Storage Capacity | Comp. Cap. Load | Storage Capacity |
| 1 | 34 | 35 | 36 | 37 | 38 | 39 | 40 | 41 | 42 | 43 | 44 | 45 | 46 | 47 |
| 17 | 1.000 | 0 | | | | | | | | | | | | |
| 18 | 0.944 | 11,340 | 1.000 | 0 | | | | | | | | | | |
| 19 | 0.895 | 21,480 | 0.947 | 11,360 | 1.000 | 0 | | | | | | | | |
| 20 | 0.850 | 30,600 | 0.900 | 21,600 | 0.950 | 11,400 | 1.000 | 0 | | | | | | |
| 21 | .810 | 38,860 | .857 | 30,870 | .905 | 21,710 | 0.952 | 11,420 | 1.000 | 0 | | | | |
| 22 | .773 | 46,370 | .818 | 39,270 | .864 | 31,100 | 0.909 | 21,820 | 0.955 | 11,470 | 1.000 | 0 | | |
| 23 | .739 | 53,220 | .783 | 46,960 | .826 | 39,650 | 0.870 | 31,300 | 0.913 | 21,920 | 0.957 | 11,480 | 1.000 | 0 |
| 24 | .708 | 59,510 | .750 | 54,000 | .792 | 47,490 | 0.833 | 40,010 | 0.875 | 31,500 | 0.917 | 21,990 | 0.958 | 11,510 |

The values in Table 1 were computed by means of the above two equations.

**Example 2**—In a chapel, cooling will be required for 6 hr approximately every third day. This 6 hr period includes the time required to cool the chapel to the required temperature. The water in the storage tank is to be allowed to rise 10 deg. Assume that the compressor is to be allowed to run for 60 hr. If the maximum cooling load is 50 tons, find: (a) The compressor capacity required. (b) The storage capacity required.

*Solution*—

a) Using the first of the two equations above,

$$\frac{Hc}{50} = \frac{6}{60}$$

$Hc = 5$ tons compressor capacity required.

The compressor should be selected for 5 tons at the *average* suction temperature.

b) Using the equation for $Hs$ given above,

$$Hs = 12{,}000 \times 6 \left(1 - \frac{6}{60}\right)$$

$= 64{,}800$ Btu per ton of cooling load.

The water storage can be figured by means of the equation published in HPAC, August, 1941, p. 502, which is,

$$\text{Gallons} = \frac{Hs}{8.33\ sc\ (t_2 - t_1)}$$

$$= \frac{64{,}800}{8.33 \times 10}$$

$= 778$ gpm per ton of cooling load.

Total storage required $= 778 \times 50 =$ 38,900 gal.

## Load Factors

Although the maximum cooling load computed for the conditioned spaces may occur occasionally, it is seldom that it will last throughout the cooling period. Thus, in the solution of Example 1, it was assumed that the maximum cooling load of 75 tons would exist during the entire cooling period of 6 hr. Usually the average conditioning load that exists throughout the cooling period is less than the maximum load. For this reason storage tanks and compressors are not usually sized for the maximum load, but rather for the average load that is expected throughout the cooling period. Thus, the compressor and storage capacity selected for Example 1 would be perfectly satisfactory for an installation in which the maximum cooling load was 100 tons and the load factor 75 per cent. This means that the average load throughout the cooling period would be 75 tons, which value was used in sizing the storage tank in Example 1.

The load factor—the ratio of the average load to the maximum load—will vary from approximately 40 per cent up to almost 100 per cent depending upon the type of installation. Few data are available on this point and the designer must use his own judgment as to the probable load factor that will be encountered on a particular installation.

After the average cooling load for which the system has been designed has been determined, it is desirable to add some allowance for heat gains in the storage tank and piping. An allowance of roughly 10 per cent will usually prove satisfactory. In other words, to the 75 ton average load mentioned in the preceding paragraph, 7.5 tons should be allowed for losses—the system should be designed for an average cooling load of 82.5 tons.

Even though the compressor and the storage tank are sized for the average load, the cooling coils or air washer, the fan delivering the conditioned air, and the pump circulating chilled water between the storage tank and the air conditioning unit should be sized for the maximum load. If they are sized in this way, the system will be able, without any difficulty, to take care of the maximum load for the short period that it usually lasts.

Where the size of the installation warrants a more elaborate analysis, it is a good plan to compute the load for each hour of the day. These hourly loads are then plotted on graph paper against time of day. The sum of the total hourly loads divided by the number of hours that the compressor will be allowed to operate gives the compressor capacity required. Then by drawing a straight line which represents the compressor capacity on the chart, it is easy to determine during which hours of the day the water in the tank will be cooled by the excess compressor capacity and when it will be heated by the excess load. The total storage capacity required can also be determined by means of such a graph.

## Load Variation

The length of time that the compressor runs is usually controlled by a thermostat which stops the compressor when the temperature of the water has been lowered to some predetermined point. If the average cooling load is, say, half of the cooling load for which the compressor was originally selected, the running time of the compressor will be exactly half of the running time for which the system was originally designed. On the other hand, in this case the change in the temperature of the water in the storage tank will not be half of the change for which the tank was originally designed. The change in the temperature of the storage water is not proportional to the running time of the compressor. The change in temperature is proportional to the *difference* between the compressor running time and the time that cooling is required, because only during this period is the storage water cooled—for the duration of the cooling load, the water is heated.

The number of hours that a compressor must run when the cooling load is less than the maximum and the change in the temperature of the stored water can also be found by means of Table 1.

**Example 3**—Take the conditions of Example 1. Assume that the average cooling load during the 6 hr period amounts to only 55 tons. The storage tank and the compressor have been sized for an average cooling load of 75 tons. Find: (a) The time that the compressor will operate. (b) The change in the temperature of the water in the tank.

*Solution*—

a)

$$\frac{\text{Compressor capacity}}{\text{Load}} = \frac{20.5}{55} = 0.373$$

In Column 12 of Table 1 for a cooling period of 6 hr, the value of (compressor capacity/load) nearest to 0.373 is 0.375. In the same row as the value of 0.375, find in Column 1 that the compressor must run slightly more than 16 hr when the average cooling load is 55 tons.

b) In Column 13, adjacent to the value of (compressor capacity/load) = 0.375, find that the energy stored = 45,000 Btu per ton of cooling load, or a total storage of 45,000 × 55 Btu. From Example 1, when the load is 75 tons, the tank temperature changes 8 deg, and under maximum conditions, the storage amounts to 52,370 Btu per ton of load, or a total storage of 52,370 × 75 Btu. Therefore, the change in temperature when the load is only 55 tons will be

$$\frac{45{,}000 \times 55}{52{,}370 \times 75} \times 8 = 5 \text{ deg. change}$$

in temperature of water in the storage tank.

The preceding example can also be solved quite simply by means of the following equations:

$$\frac{H_c}{H_L} = \frac{\theta_L}{\theta_c}$$

$$\frac{H_{s2}}{H_{s1}} = \frac{\theta_{c2} - \theta_L}{\theta_{c1} - \theta_L}$$

The use of these equations eliminates the need of interpolation in the table and gives somewhat more precise results for the occasions when they are needed; the second of the above equations is also useful for those cases where cooling is not required every day, as in the chapel of Example 2.

**Example 4**—Solve Example 3 by means of the two equations given above.

*Solution*—(a) Using the first equation,

$$\theta_c = 6 \left( \frac{55}{20.5} \right) = 16.1 \text{ hr.}$$

b) Using the second equation,

$$\frac{H_{s2}}{H_{s1}} = \frac{16.1 - 6}{22 - 6} = 0.63$$

Temperature rise of water in storage tank = 0.63 × 8 = 5 deg.

## Running Time

The compressor capacity and size of storage tank are shown in Fig. 1

for various selected running times of the compressor for the conditions of Example 1. Notice that if the operating time of the compressor had been selected for 24 hr instead of 22 hr as in Example 1, the required capacity of the compressor would have been reduced by less than 2 tons, and the required storage capacity would have been increased by less than 2000 gal. The reduction in compressor capacity for an increase in compressor running time from 22 to 24 hr is generally small. As is shown in the following paragraphs, a margin of safety is obtained by designing the system for a compressor running time of less than 24 hr. Where the smallest practical size of compressor is desired, a compressor running time of 22 or even 20 hr should be selected rather than a running time of 24 hr. Any running time beyond the period of approximately 22 hr does not usually reduce the required refrigerating capacity sufficiently to warrant neglecting the advantage of the small margin of safety obtained.

By means of the following equation—derived from the equation $H_C/H_L = \theta_L/\theta_C$ for constant compressor capacity and duration of cooling load—it can be shown for the conditions of Example 1, that if the compressor runs 24 hr instead of 22 hr, the storage system will be able to take care of an average conditioning load approximately 9 per cent greater than the load for which the tank was originally designed.

$$\frac{H_{L2}}{H_{L1}} = \frac{\theta_{C2}}{\theta_{C1}}$$

$$\frac{H_{L2}}{H_{L1}} = \frac{24}{22} = 1.09$$

If the compressor runs 24 hr instead of 22 hr, the equation $[H_{S_2}/H_{S_1} = (\theta_{C_2} - \theta_L)/(\theta_{C_1} - \theta_L)]$ can be used to show that the storage capacity will be increased by 12½ per cent. In other words, the temperature rise of the water in the storage tank, instead of being 8 deg as assumed in Example 1, will be 12½ per cent greater than 8 or 9 deg. Obviously if the water is heated 9 deg by the greater cooling load, the compressor must run longer to cool the water back to the initial temperature of 40 F as given in Example 1. It may seem odd that an increase in storage capacity of 12½ per cent can handle an increased cooling load of only 9 per cent. The reason for this is, of course, that the actual cooling load is absorbed not by the storage tank alone, but by the storage tank and compressor together.

From the foregoing it is apparent that the advantages in selecting a 24 hr running period are usually not very great and that a 22 hr running period is desirable not only because it provides some margin of safety, but also because it allows a few hours for repairs in case of minor emergencies.

*[Next month, Mr. Goodman will discuss piping and storage tanks, control of compressor, and motor overloading, and will give derivations of his equations.]*

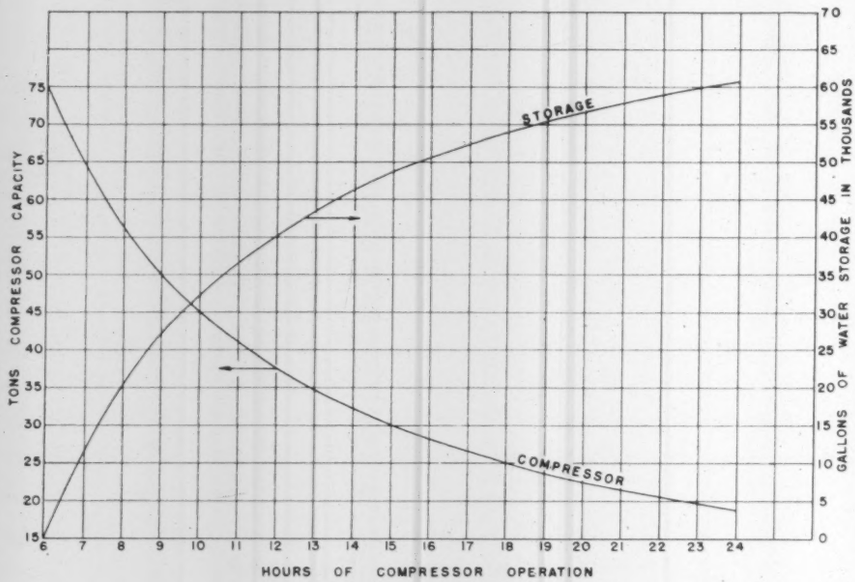



**Fig. 1—Required compressor and storage capacities for various operating periods of compressor for conditions of Example 1**

# INTERCHANGE OF POWER AND STEAM

THE EDITOR—

I have read Mr. Ross' article on interchange of power and steam, published in the September HPAC, and believe he has told the story very well.

I think it will be of interest to your readers to know that the operation of this high pressure steam line has been so satisfactory over the first eight years of its life that in 1940 we contracted with the power company for a continuation of the contract for a period of 12 years. Naturally we would not have done this if we had not received an uninterrupted supply of both steam and electricity on a basis of dependability equal to what we could have done with a plant entirely under our control.

There are inherent savings in an arrangement of this kind which can be divided between a utility and a consumer if they will approach the whole subject in a spirit of fairness and a realization of the other man's problem.—W. A. HANLEY, Eli Lilly and Co. Member of HPAC's board of consulting and contributing editors.

# Conservation for National Defense

# How to Save Heating Fuel

## E. A. Tyler, Rental Manager, Tells of Steps Taken to Save Over 40 Per Cent of His Building's Steam

IN THIS national emergency—when fuel means power, and power is the means to production of the wherewithal to defend ourselves against aggressors—we that control the designing and construction of fuel consuming equipment and the management of property should put our shoulders to the wheel and eliminate as far as possible waste of fuel.

### Dental Revery

Sometimes it takes a great catastrophe to jar men out of their lethargy and sometimes a minor incident serves the purpose as well. As I sat in my dentist's chair on a sub-zero day in 1934, with the drill in my tooth feeling about one inch in size, I looked out on the west side of the beautiful limestone facade of the First National Bank building in St. Paul—which building I had a major hand in planning, building and operating. In an effort to take my mind off the painful work the dentist was doing, I started counting the windows; coming to the 27th floor I got a jolt that embarrassed the dentist. Two of these windows were wide open, and that meant waste so obviously that I immediately investigated when I returned to the office. Not only were these two windows wide open, but dozens of windows in unoccupied and unfinished space were open from a crack to an inch.

That incident was the jolt I needed to shake me out of the belief that our steam consumption was down to bedrock. This belief was engendered by the fact that when we constructed the building in 1930-31, we had purchased and installed what we believed to be one of the best systems of heating control on the market, and had kept the engineer who supervised the installation on the job for a full heating season to thoroughly train our boiler room personnel in its operation; the first year's operation came up exactly to the guarantee. We thought this equipment was doing a great job—and it was, for it easily paid for itself the first three years.

### Degree Days

I would like to meet and shake the hand of the man that invented the degree day method of measuring heating requirements, for in my opinion the building owners, operating and mechanical engineers should erect a monument to him. It was about the time of the above mentioned incident that I commenced to find out about degree days, and that is when I really discovered what we were doing with our steam consumption. With this new measuring rod, I could plot it from day-to-day, month-to-month, and year-to-year. I plotted it back to the beginning, setting it up in graphs showing daily consumption for each zone, as well as the total. For the first time I really had a picture of how skillfully our men were operating the heating plant.

A year's operation looked like a profile of the Rocky Mountains, and the valleys and peaks in the spring and fall would defy any mountain climber that ever lived. Surely, I reasoned, if the plant could be operated at 5000 lb of steam per degree day for one day, it should be somewhere near that the succeeding day—but it wasn't. The next day it would probably be up to 6000 or even 8000 lb. Then is when I went to work in earnest to find out if that line could be leveled out—or almost leveled.

### Five Phases

The problem of getting started on a steam conservation program presented five phases: (1) Stopping the leaks of perfectly good heated

**HOW TO SAVE the fuel required for heating—essential service in plants, commercial buildings, institutions and all structures—is a vital problem to operating men always. During our national effort in defense production, it is even more important and becomes a patriotic duty. . . . Mr. Tyler, rental manager of the First National Bank building, St. Paul, Minn., explains in detail here the initial steps taken to reduce the steam consumption for heating this large building. This conservation program has resulted in a saving of over 40 per cent during the period of the last few years.**

air; (2) balancing the radiation and determining the proper time for night shut-off; (3) improvement and simplification of control; (4) improvements in the skill of the operating personnel; and (5) a simple system of accounting that would bring to my attention daily, monthly and annually, sufficient information to the end that I could immediately detect any deviation from an efficient standard of operation. I did not see it quite that plain in the beginning, but that is the way it worked out.

The first two phases will be discussed in this article; the last three, and the results obtained, will be outlined in a second one.

## Stopping the Leaks

A high building is like a chimney—in the winter it is warm inside and cold outside. In the lower 60 per cent of its height the inward pressure of the outside air will be in direct proportion to the outside temperature. In the upper 40 per cent, the inside air will be under pressure. Any openings below the 60 per cent line will admit cold air in direct proportion to the area of the opening and the outside temperature, plus wind pressure on the windward side of the building. Any openings above the 60 per cent line will make it that much easier for the outside air to come in below by relieving the pressure on the interior of the building. The thing to do is to stop dead all openings both above and below. But what about ventilation? you may ask. It is true you have to have ventilation for the occupants, and I am certain that at temperatures anywhere below 20 F for most types of occupancy below the 60 per cent line, infiltration through the best weatherstripped windows will provide adequate ventilation, and above the 60 per cent line the occupants will get a nice outward draft if they crack their windows. For mass occupancy, mechanical ventilation should, of course, be provided. We found a great many leaks that we stopped, and that helped materially in reducing our steam consumption.

Connected to our new building is an older 16 story building. We must have been losing 10,000 cfm of warm air through the top of this building as there were some 12 skylights, all with large openings and pipe radiation encircling the light wells. We closed up as many of these skylights as we could and removed the radiation. The penthouse had two iron plate doors leading to the roof, both of which had warped badly and let a gale of warm air through when the temperature was down. Several of the hollow metal windows in the penthouse had holes in the wire glass, which were repaired. The elevators are on the north side of the building and the designer had put in 18 hollow metal windows to provide light for this elevator shaft. It was practically impossible to clean these windows, so in times past they had

been painted dark brown to improve their appearance. You could throw a cat through most of these windows, and it was difficult to keep the corridors and elevator shafts warm, even though there were great banks of radiation in the shafts and corridors. We did not even take the trouble to remove these windows—we simply filled the reveals full of rock wool, did a neat job of covering them with plywood nailed to frames and painted to match the walls. This stopped a big leak, and incidentally we now have comfortable elevator shafts and corridors. From the appearance inside you would never know there were any windows there.

The ground floor corridor is connected to a 300 car garage, with very active traffic. When the new building was built we felt that two swing doors at each entrance to the garage gave ample protection. Frequently both swing doors and the garage door would be opened at the same time, which let in a terrific blast of cold air, cooling off the ground floor lobby and pushing the warm air out the top of the building. There was no room to install revolving doors at these openings, but we solved that by an obvious but neat trick; we put in a third door, designed so that it could be taken out in the summer.

The building code requires steel windows on the sides of the building exposed to other buildings. The new part has modern steel sash, while the old part has old fashioned hollow metal windows. Both the old and new leaked so badly that an unbalanced condition was created which added unnecessary infiltration. We induced a metal window manufacturer to design and build special lightweight steel double hung frames and sash, and installed them on the inside. These worked perfectly.

The banking areas of the building are air conditioned, with outside air openings totaling 150 sq ft situated at the 5th and 9th floor levels, closed with ordinary iron dampers. Investigation indicated that these openings passed almost as much air when the fans were not running in the winter as they did when the fans were running, consuming an estimated 1,800,000 lb of steam annually. This leak was stopped by the installation of auxiliary wood dampers. (For a detailed description see p. 507, August, 1941, HPAC).

An old two story brick and frame building is connected on the ground floor to the two main buildings by an arcade. The suction set up on this building by being connected to the higher buildings made it almost impossible to heat the lower building, and its steam consumption was way out of line. We stopped this leak by tightening all windows and putting weatherstripping in first-class condition, installing storm doors on all openings possible, and insulating the ceiling of the second story under the roof with 4 in. of rock wool. These improvements resulted in a reduction of one-third in steam consumption for that zone.

**The Chief, Lewis Hohenthaner, checks inside and outside temperatures with a resistance thermometer installation at the First National Bank building, St. Paul. Temperature information is of course essential to the economical and proper operation of any large heating system**

The janitors who service the occupied areas are required to check and tightly lock each window. A little snooping at night after the men had finished their work revealed that there were many omissions of this chore. A little persuasion and checking by the night superintendent reduced this loss to a minimum.

If we could see through our blind spots I am sure there are other leaks similar to the foregoing that could be found and eliminated.

## Night Shut-Off

In my travels I have been in a very large number of office buildings and hotels, mostly in the northern part of the United States, and I am sure that it would not be wide of the mark to say that 75 per cent of them overheat. Probably most cases of overheating are caused by unbalanced radiation, and windows and doors that admit more air than the engineer allowed for when the heating plant was laid out. The first requirement of an economical heating plant is a near perfect balance in feed lines and radiation, which I well know is elemental. But I also know that it is practically impossible for an engineer to compute feed line and radiation requirements for a large building with hundreds of rooms, in the blueprint stage, without making spot errors that will run anywhere from 5 to 10 per cent. It should be the operating engineer's job to check these errors and correct them.

There are still some operating engineers who believe there is no economy in shutting off the steam at night, and there are many buildings where this is true. However, in office and loft buildings, and other buildings that have principally a daytime occupancy the great majority now agree that shutting off at night is a decided economy. We had been shutting off at night, but the schedules used had been based on guesswork, and in many cases had been left to the judgment of the night engineer.

The problem was to check the balance of the radiation and locate excessive infiltration, and to establish accurate off-and-on schedules for night operation. To accomplish this, I first purchased a good portable recording thermometer and a supply of paper graph faces. Then I established a tentative off-and-on schedule for night operation, at the same time giving instructions to the night engineer that he was to make frequent trips through the building and if he should find that the temperature in any area was dropping close to 60 F, he should give that zone a shot of steam. During the months of November, December, January and April I placed the recording thermometer in different places each 24 hr period (some typical, and many suspected of an unbalanced condition), noting the date, location, high, low and mean temperature on each graph. During this four month period we established by trial and error the most efficient night shut-off periods and located most of the unbalanced conditions. The shut-off periods were finally set up in the form of a chart for each zone that shows at a glance the time to shut off and turn on the steam at any given outside temperature.

The correction of the unbalanced radiation was taken care of as located, and strange as it may seem, we found several unbalanced conditions due to too small risers and leaky expansion joints. In the older buildings we checked all mains and risers for leaks—and found plenty. Ours is a two pipe vacuum job,

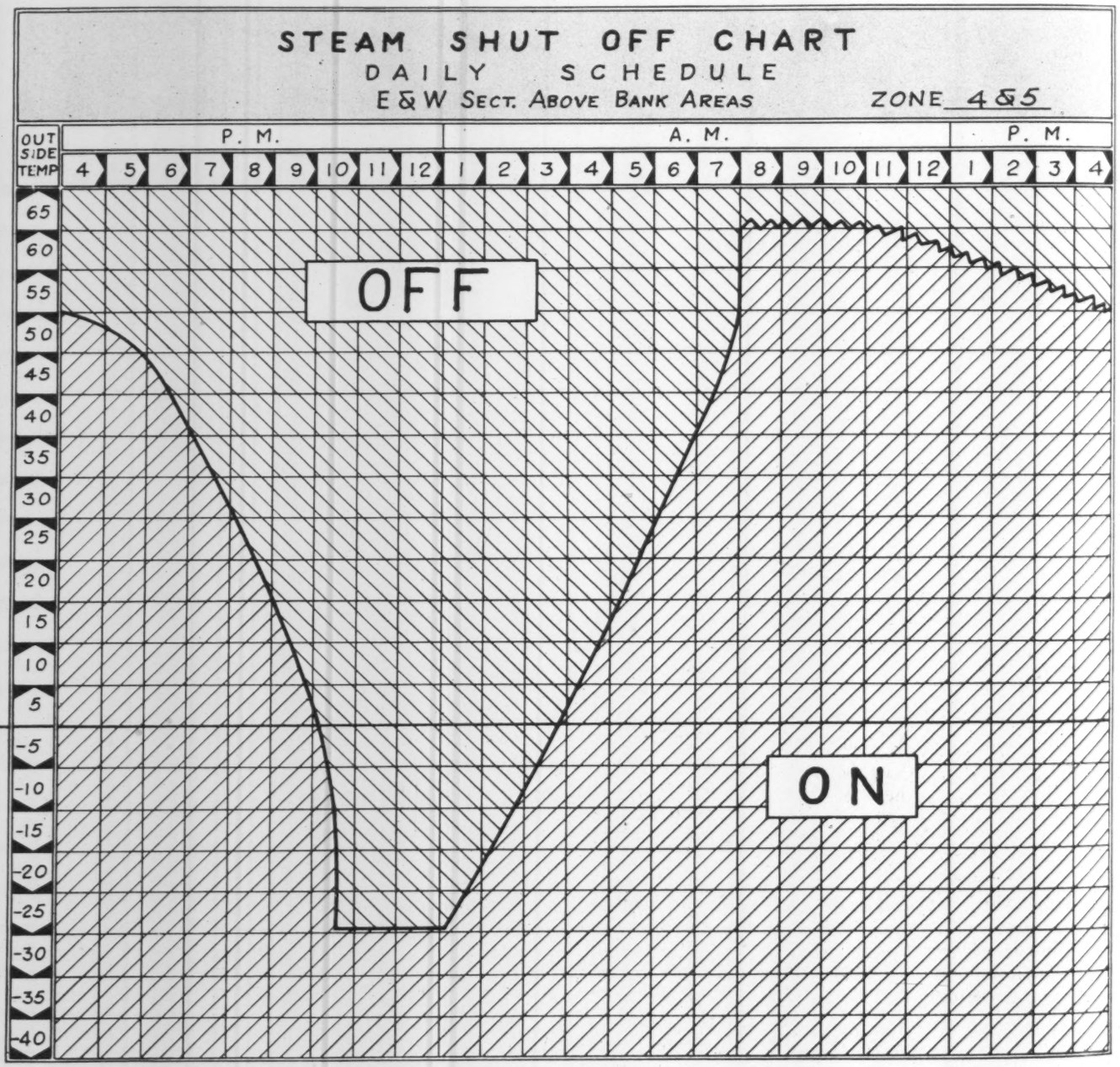
running vacuums as high as 18 in. in very mild weather, and even a small leak in a riser will bring the tenants "hollering for heat."

## Riser Too Small

The northeast corner of the older building was unbalanced and we could not cure it. We added radiation until we were ashamed of the appearance of the rooms, but we still had to feed the zone more steam than was needed to keep this corner warm. The riser was covered up so that it could not be seen and the plans of the building were not kept when the building was finished, so we were not able to check the riser size—neither were we able to get the plans from the architect.

Finally, in desperation, we found the contractor who had installed the plant, induced him to dig out from a mess of old plans a very much battered and torn set of the original heating plans which dimly showed this troublesome northeast corner. A glance quickly showed the cause of the trouble. The original plans had provided one large hollow metal window in the northeast corner room, but sometime during the period of construction another identical window was added to this room and necessary radiation was added to compensate for this window—but the riser had been left the same size! A few calculations quickly showed that the riser was much too small to carry enough steam.

To remove the riser and install a larger one would be an extremely costly job. How could this be cured? Our able steamfitting contractor came through with the answer. By checking the size of the adjoining riser he found that it was about 35 per cent larger than was needed. By tying three 1¼ in. cross-overs from the riser that was too large to the riser that was too small, at the 12th, 8th and 4th floor levels, adequate supply was established for our last unbalanced condition and at a fraction of the cost of putting in a new and larger riser.


STEAM SHUT OFF CHART
DAILY SCHEDULE
E & W SECT. ABOVE BANK AREAS
ZONE 4&5
OUT SIDE TEMP
P. M.
A. M.
P. M.
4 5 6 7 8 9 10 11 12 1 2 3 4 5 6 7 8 9 10 11 12 1 2 3 4
65 60 55 50 45 40 35 30 25 20 15 10 5 0 -5 -10 -15 -20 -25 -30 -35 -40
OFF
ON

Heating · Piping and Air Conditioning

# AMERICAN SOCIETY of HEATING and VENTILATING ENGINEERS

## Journal Section

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WESTERN NEW YORK: *Organized,* 1919. *Headquarters,* Buffalo, N. Y. *Meets,* Second Monday. *President,* W. R. Heath, 119 Wingate Ave. *Secretary,* Herman Seelbach, Jr., 120 Mariner St.

WISCONSIN: *Organized,* 1922. *Headquarters,* Milwaukee, Wis. *Meets,* Third Monday. *President,* T. M. Hughey, 906 N. 4th St. *Secretary,* O. A. Trostel, Rte. 2, Thiensville, Wis.

# Pressure Loss Caused by Elbows in 8-Inch Round Ventilating Duct

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Bethlehem, Pa.

This paper is the result of research sponsored by the American Society of Heating and Ventilating Engineers in cooperation with Lehigh University

## Introduction

A LARGE per cent of the total pressure loss experienced in ventilating systems is caused by elbows. If this loss can be reduced, more efficient systems may be designed. In 1939 the American Society of Heating and Ventilating Engineers entered into a cooperative agreement with Lehigh University for the purpose of studying the pressure loss due to elbows. The program will include the study of all types of elbows for 8-in. round and 7 x 7 in. square and other size ducts. This paper presents the study of round elbows in 8-in. duct. The 7 x 7 in. square duct is now under investigation.

## Pressure Drop in Straight Duct

Apparatus for the study of pressure drop in straight duct was set up as shown in Fig. 1A. Duct having 40 joints per 100 ft was used with all joints taped to prevent leakage. Air supplied by a fan having a capacity of 3000 cfm at the pressures desired, discharged into a box which broke up the circulation caused by the fan so that normal turbulence was maintained in the test section. An egg crate straightener was used to further insure normal turbulent flow. The duct discharged into a 30-in. diameter plenum chamber fitted with a calibrated discharge nozzle. Static tube pressure taps were placed every 10 ft along the test section. The reasons for the use of the static tube may be found in Appendix I. These taps were connected differentially through commercial single inclined tube manometers. The last pressure tap before the plenum chamber was connected to a manometer having the other leg open to the atmosphere. An over-all pressure differential was taken to be used as a check on the small differentials.

The velocity was obtained by dividing the quantity of air flowing by the area of the duct. All calculations were based on an assumed nominal diameter of 8-in. for the duct. This is not the exact diameter since so-called 8-in. duct has a diameter which varies from $7\frac{7}{8}$ in. to $8\frac{1}{16}$ in. For this reason the velocity used was an average nominal velocity. (See Appendix III for derivation of velocity equation.) The quantity of flow was varied by throttling the inlet of the fan.

The results of the investigation on pressure loss for 100 ft of straight duct are shown in Fig. 3 by the curve labeled *Straight Duct*. The straight duct pressure loss is being investigated by the Society's Pittsburgh Laboratory, but, as will be shown later, these values are needed in the present investigation of elbow pressure loss. The results[1] of this investigation check with those of the Laboratory as presented in the paper, previously published.

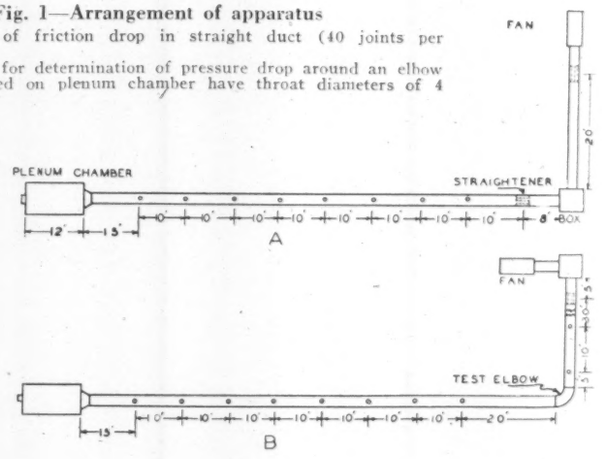
## Pressure Drop in a Single Elbow

Apparatus for the study of pressure drop in a single elbow was that used in the straight duct test but modified as shown in Fig. 1B. An elbow is defined as the section of duct joining two straight lengths of duct. Three pressure taps preceding the elbow and three following were used. At various times during the test additional taps were used as a check on the work.

**Fig. 1—Arrangement of apparatus**

*a.* For study of friction drop in straight duct (40 joints per 100 ft)

*b.* Test set-up for determination of pressure drop around an elbow (nozzles used on plenum chamber have throat diameters of 4 and 8 in.).




The results of the investigation of pressure loss in a single elbow are shown in Fig. 2, a plot of pressure drop *vs.* length along the duct for different velocities. Pressure loss in an elbow is caused by three flow phenomena: (1) the friction of the air particles against the duct wall; (2) the loss due to turbulent flow; and (3) the loss due to a change in direction of flow. The first two items constitute the loss experienced in straight duct and also occur in an elbow because it

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[1] Analysis of the Factors Affecting Duct Friction, by J. B. Schmieler, F. C. Houghten and H. T. Olson. ASHVE Journal Section, *Heating, Piping and Air Conditioning*, March, 1940.

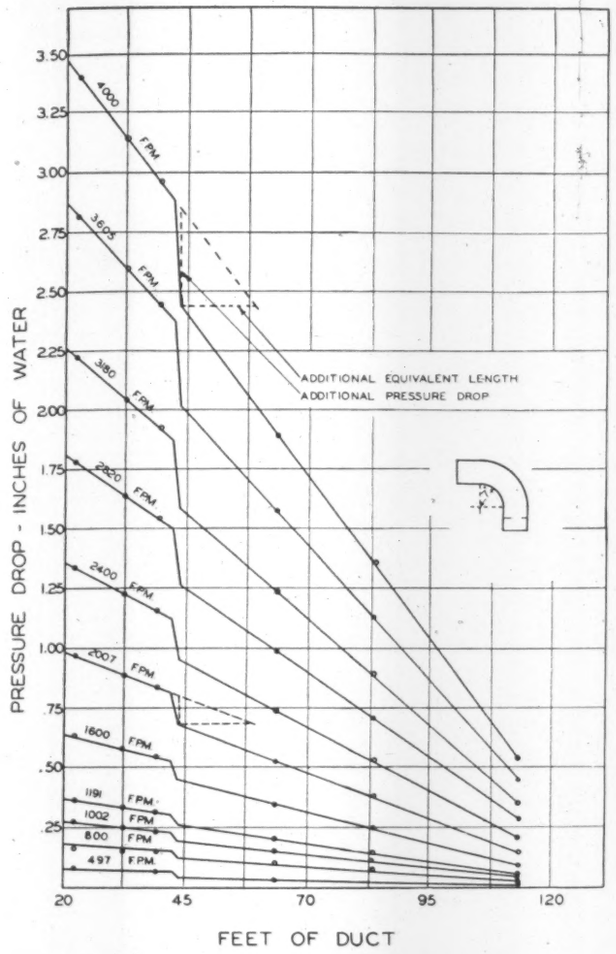


Fig. 2—Relation between pressure drop and velocity for a 7-in. inside radius elbow (note that additional equivalent length does not change with velocity)

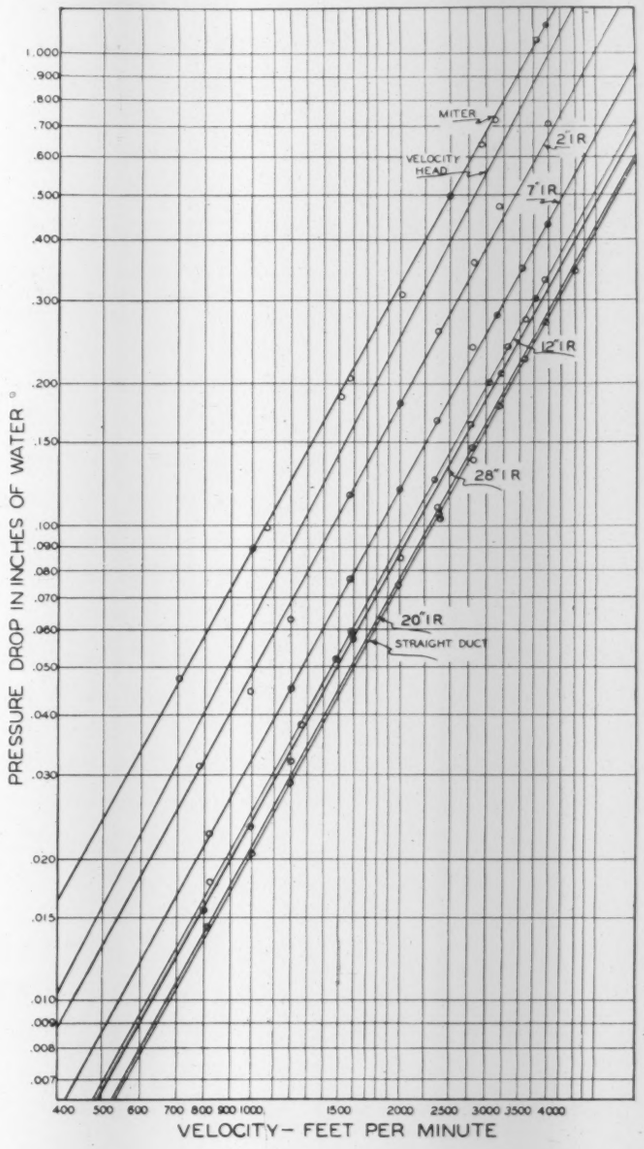


Fig. 3 (right)—Relation between total pressure drop and velocity for all elbows tested. The curve, straight duct, is for 10 ft of duct with 40 slip joints per 100 ft

has a definite length. The third loss is unique to the elbow, and may be termed the *additional pressure loss caused by the elbow. This additional loss is determined by finding the excess of pressure drop for a section of duct including an elbow over the calculated pressure drop for a section of straight duct having an equal centerline length.* The total loss caused by the elbow is made up of the friction loss based on its centerline length and the additional loss determined by the method described herein.

Examination of Fig. 2 will show that the pressure gradients preceding and following the elbow are parallel straight lines but some distance apart. If the length of duct involved were straight and the elbow omitted, points on a pressure gradient would be on the same straight line. Because of the abnormally high pressure drop in the elbow with respect to its length, a line indicating the pressure drop along the center line length of the duct and elbow shows a discontinuity. The vertical distance on the pressure axis between the two segments of the pressure gradient is equal to the additional pressure loss due to the elbow, and the horizontal distance along the duct length axis between the segments is precisely that length of straight duct which might have been added to the elbow length to cause an equal pressure drop. Since there is within the elbow itself a friction drop admittedly proportional to the centerline length of the elbow, we may add this friction loss to the additional loss and term the result the total loss due to the elbow. Similarly, the length of the elbow added to the additional equivalent length may be termed the total equivalent length.

Fig. 2 also shows that for a given elbow the additional equivalent length does not change with velocity, consequently the total equivalent length will not change. For a given elbow a unique length of straight duct may be substituted causing a pressure loss exactly equal to the loss due to the elbow. It immediately follows that the loss in an elbow is the same power function of velocity as the loss in straight ducts. Equivalent length of duct is of more significance than pressure loss because as was shown it is independent of velocity. Additional equivalent length, the excess of pressure drop for a section of duct including an elbow over the calculated pressure drop for an equiva-

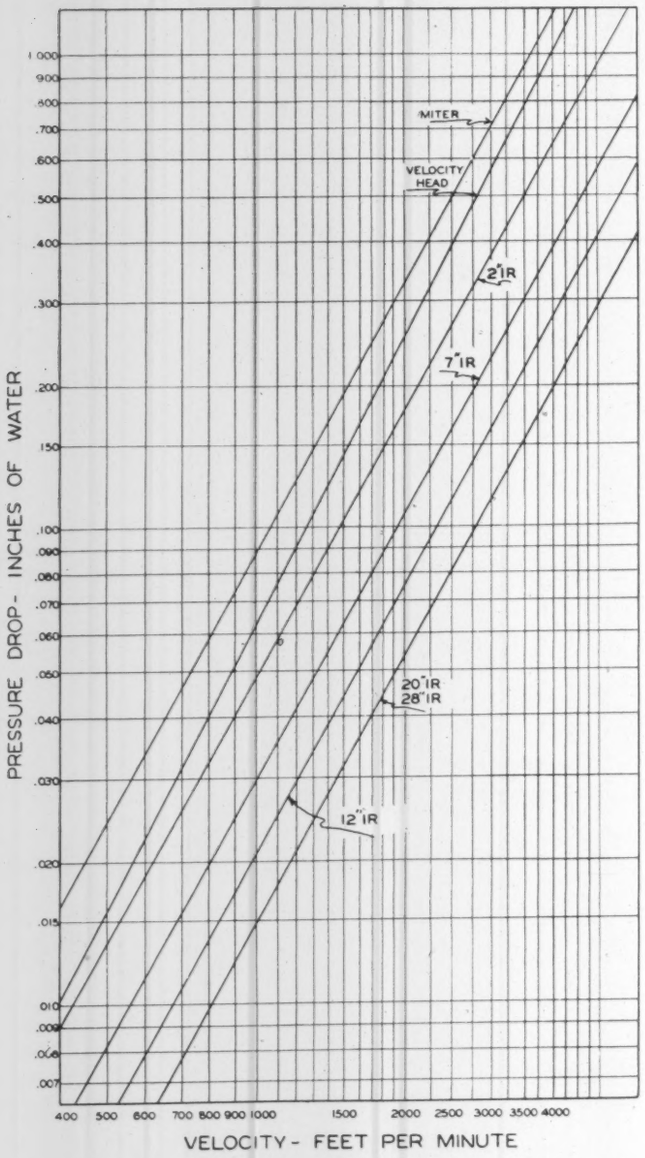


**Fig. 4—Relation between additional pressure drop and velocity for all elbows tested**

Additional pressure loss is defined as the excess of measured pressure drop for a section of duct including an elbow over the calculated pressure drop for a section of straight duct having an equal centerline length.

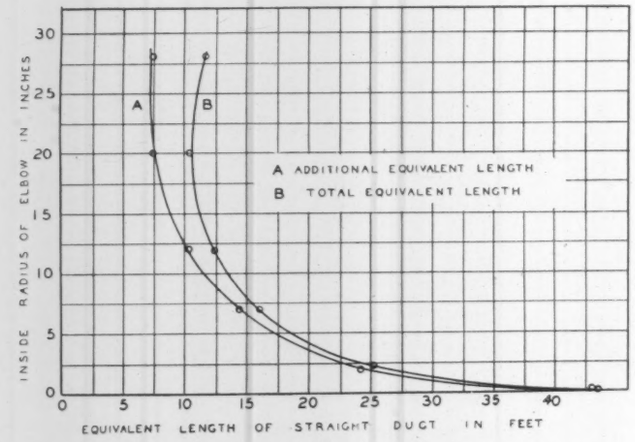
INSIDE RADIUS OF ELBOW IN INCHES
EQUIVALENT LENGTH OF STRAIGHT DUCT IN FEET
A ADDITIONAL EQUIVALENT LENGTH
B TOTAL EQUIVALENT LENGTH

**Fig. 5—Relation between inside radius of elbows and equivalent length of straight duct at any velocity**

lent centerline length of duct divided by the pressure loss per foot of duct, is for some purposes of more importance than total equivalent length.

## Effect of Varying Radii on Pressure Loss in Elbows

Elbows with radii from 0 to 28 in., shown in Fig. 6, were investigated in the arrangement shown in Fig. 1B. The procedure for measuring the pressure loss and velocity already described was used for all elbows. The curves of total loss *vs.* velocity are given in Fig. 3. A plot of additional loss *vs.* velocity is given in Fig. 4. Examination of the curves will show that the total pressure loss decreases with increasing radii. When an optimum value has been reached, further increases in radii will increase the total pressure loss. The additional pressure loss decreases with increasing radii. It may be concluded that as the radius increases the friction loss becomes the dominant value.

The numerical relationship existing between elbows of different radii and equivalent lengths of straight duct is shown in Fig. 5. The additional equivalent length of elbows, Curve *A*, of Fig. 5, decreases with increasing radii and very nearly approaches a constant value at large radii. Also in Fig. 5 Curve *B* of the total equivalent length of straight duct reaches a minimum value and then increases with increasing radii.

From the results the important conclusion reached is that in elbows of large radii, the pressure loss caused by changing the direction of flow is of minor importance compared to the pressure loss caused by friction and turbulence over its centerline length.

## Elbows in Series

Two elbows separated by a length of straight duct 40 diameters long were investigated to determine the effect of one elbow on another farther down the duct. No effect of the first elbow on the second elbow was observed; therefore, it may be concluded that if the distance separating the elbows is greater than 40 diameters one elbow will not affect another.

An investigation was made, using the set-up shown in Fig. 7A to de-

termine the loss of 180 deg bends made up of two 90 deg elbows. The bends, shown in Fig. 8 were tested in the same apparatus used previously. As a result of this study it may be concluded that the total pressure loss in 180 deg bends is never as much as that of two elbows and decreases as the radius increases. In fact for large radii the loss in 180 deg bends is scarcely more than that in the 90 deg elbow.

The same effect was noted when the elbows, installed as offsets, shown in Fig. 9, were tested as arranged in Fig. 7B. From these studies it may be concluded that after the direction of flow has once been changed, further change in direction of flow does not increase the loss materially. The one exception to this is the miter elbow. When two of these are used together, the total loss is twice that of one.

### Vaned Elbows

In the 8-in. duct two elbows, the miter and the 2 in. inside radius elbows, were fitted with shop made splitters fastened to a threaded rod extending from the inside to the outside of the duct 45 deg around the bend. The rods were movable so that the splitters could be located any place across the elbow. Splitters[2] of different heights and lengths were tried in various positions across the duct until the most effective splitter and most efficient position for it were found. Tests were then made to determine the pressure loss, the same method being used as was employed in determining the pressure loss of a single elbow.

It was found that one splitter, 4½ in. long, 7 in. high, bent on a 3½ in. radius, and located 1½ in. from the inside wall of the 2 in. inside radius elbow, gave the lowest pressure drop and a reduction of 20 per cent in the total equivalent length. A reduction of 40 per cent in total equivalent length of the miter elbow was obtained by one splitter 6½ in. long, 7¼ in. high, bent on a 4 in. radius, and located 2¼ in. from the inside wall. No appreciable decrease in loss was noted when the splitters were located near the outside wall of the elbows. No material reduction in pressure loss was obtained by the use of two splitters instead of one in the miter elbow.

[2]See Appendix IV for description of splitters tested.

### Conclusions

This paper presents a method of determining experimentally the pressure loss caused by elbows in ducts of ventilating systems.

The data presented include results of the investigation of pressure loss caused by various types of elbows in 8-in. round duct.

Significant conclusions arrived at from the data presented include the following:

1. The pressure loss caused by elbows is proportional to the 1.84 power of the velocity.
2. The pressure loss of an elbow at any velocity may be expressed by a unique equivalent length of straight duct.
3. Splitters reduce the pressure loss in elbows of small radii. One splitter in a 2-in. inside radius elbow reduces the loss 20 per cent; one in the miter elbow reduces the loss 40 per cent.
4. Equivalent length of pipe for elbows having varying radii as follows:

| Inside Radius | Additional Equiv. Length in Feet | Centerline Length in Feet | Total Equivalent Length in Feet |
|---|---|---|---|
| 0 Miter | 42.9 | 0.6 | 43.5 |
| 2 in. | 24.1 | 0.8 | 24.9 |
| 7 in. | 14.5 | 1.5 | 16.0 |
| 12 in. | 10.3 | 2.1 | 12.4 |
| 20 in. | 7.5 | 3.1 | 10.6 |
| 28 in. | 7.2 | 4.2 | 11.4 |
| 0 Miter (vaned) | 25.7 | 0.6 | 26.3 |
| 2 in. (vaned) | 19.3 | 0.8 | 20.1 |

## Appendix I

### Discussion of Static Pressure Differential Measurements

In order to select the most suitable means of measuring static pressure differential, four methods were tested, *viz*: piezometer ring, single impact tube, averaging impact tube, and single static tube.

First, piezometer rings were spaced every 10 ft and connected differentially. When the results were plotted it was found that the pressure gradient did not satisfactorily fulfill the condition of uniform slope. This deficiency may have been caused by local conditions at the piezometer ring due to the proximity of the ring to the duct joint.

Tests were next made to study the possibility of the use of impact tube, averaging impact tube, and static tube. For this study a straight section of duct, 80 ft long, was set up as shown in Fig. 10 with three surveying stations 15 ft apart. Each station was equipped with

Fig. 6—All of the elbows studied

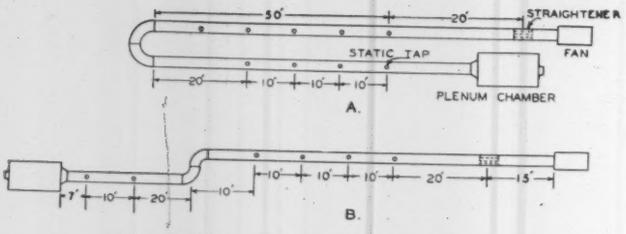


**Fig. 7—Test arrangement of apparatus**

*a.* For the study of pressure loss caused by a 180 deg change in the direction of flow

*b.* Arrangement of apparatus used in the study of offsets (nozzles used on plenum chamber have throat diameters of 4 and 8 in.)

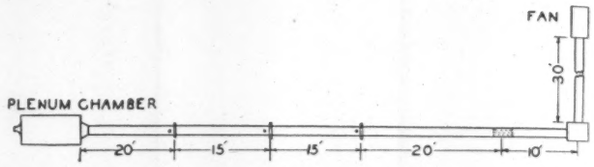


**Fig. 10—Test apparatus for the comparison of pressure measuring devices (the nozzle of plenum chamber has an 8-in. throat)**

holes, to permit insertion of survey tubes, and a piezometer ring so located that local conditions could not affect its use. Three different survey tubes, each $\frac{3}{16}$ in. inside diameter, were used. No. 1, the impact tube, had an upstream length of 4 in. terminating in a well-rounded entrance. No. 2, the averaging impact tube, was made by drilling nine 0.04-in. holes ¼ in. apart in a straight line over a 2-in. length in the middle of a tube 20 in. long and closed at one end. When used the tube extended across the duct at its diameter with the holes pointed upstream parallel to the direction of flow. No. 3, the static tube, ASHVE type, had an upstream length of 4 in. with four 0.04-in. holes drilled at right angles to each other 1½ in. from the rounded end. In making surveys, the tubes were connected differentially through a manometer to the piezometer ring.

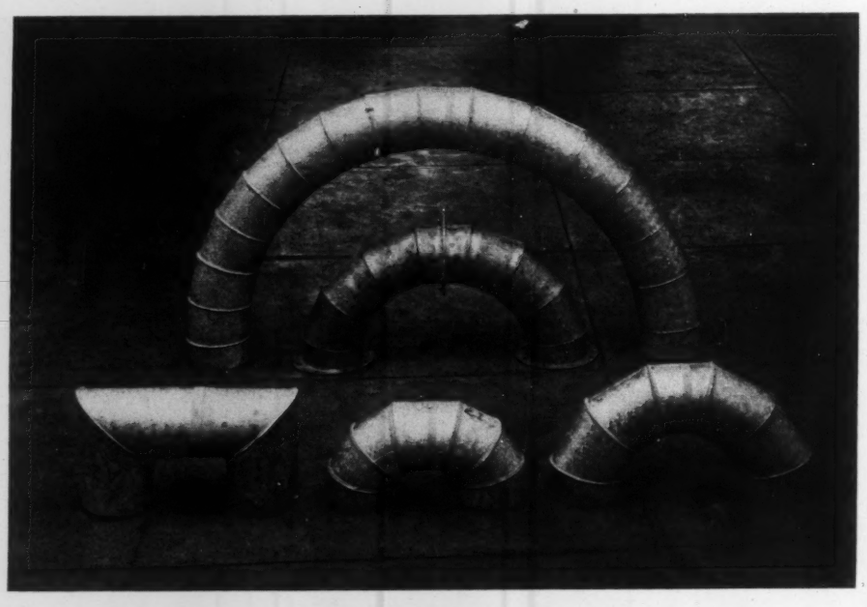

**Fig. 8—U bends made up of two 90 deg elbows of the same inside radius**

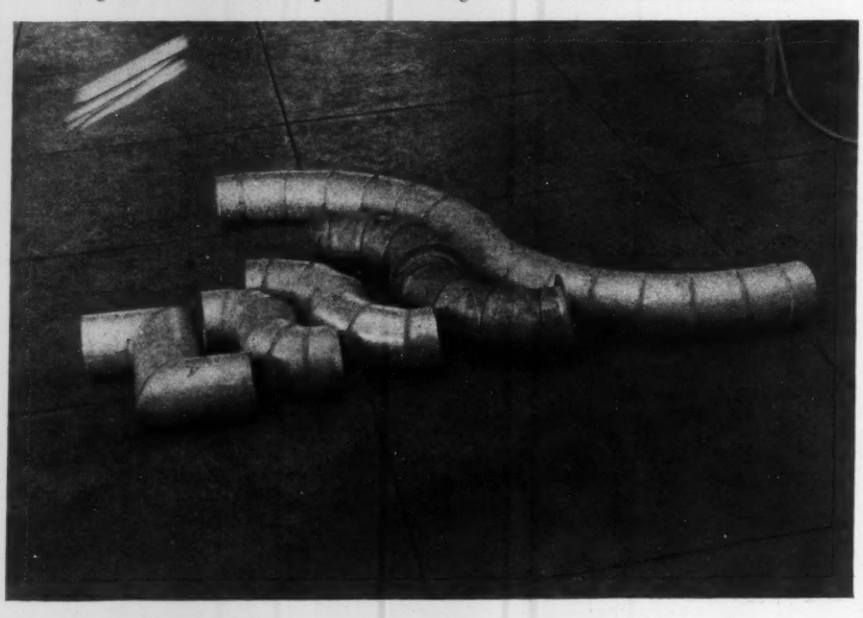

**Fig. 9—The same elbows as shown in Fig. 8 but assembled to form offsets**

The surveys were made at all three stations at velocities of 3600, 3000, 1000, and 500 fpm. In all, about 300 readings were taken. The results are shown in Figs. 11, 12, and 13. Examination of Fig. 11 will show that the impact tube does not give satisfactory results when used as a fixed instrument. Fig. 12 shows that the averaging impact tube gives more uniform pressure patterns but is still unsatisfactory except at very low velocities. Fig. 13 showing static pressure versus piezometer ring static pressure indicates that neither velocity nor location along the duct affects the result.

The preceding discussion indicates that the static tube gives the most reliable results since it is not affected by location. For this reason and because of its simplicity in installation and use, static tubes fixed at the geometric center of the duct were used to make all pressure measurements made in this investigation.

## Appendix II

### The Lehigh Micro-manometer

To insure accurate results, a calibrating manometer, Fig. 14, was developed and constructed. This manometer is patterned after one made by Professor Merriam of Worcester Polytechnic Institute. The manometer consists of two 3½-in. diameter glass cylinders ground to the same inside diameter

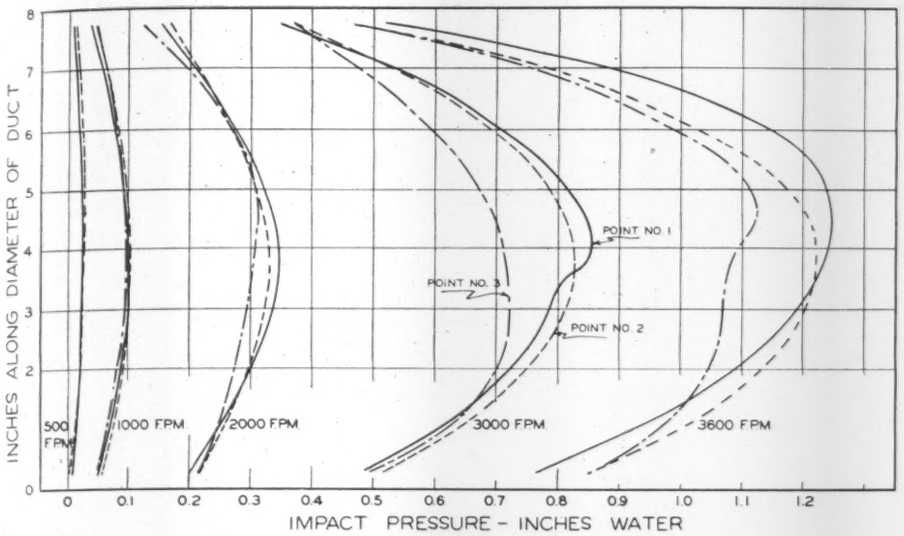


Fig. 11—Relation between position along diameter of duct and impact pressure, as measured by a ⅛ in. bore impact tube, for different velocities

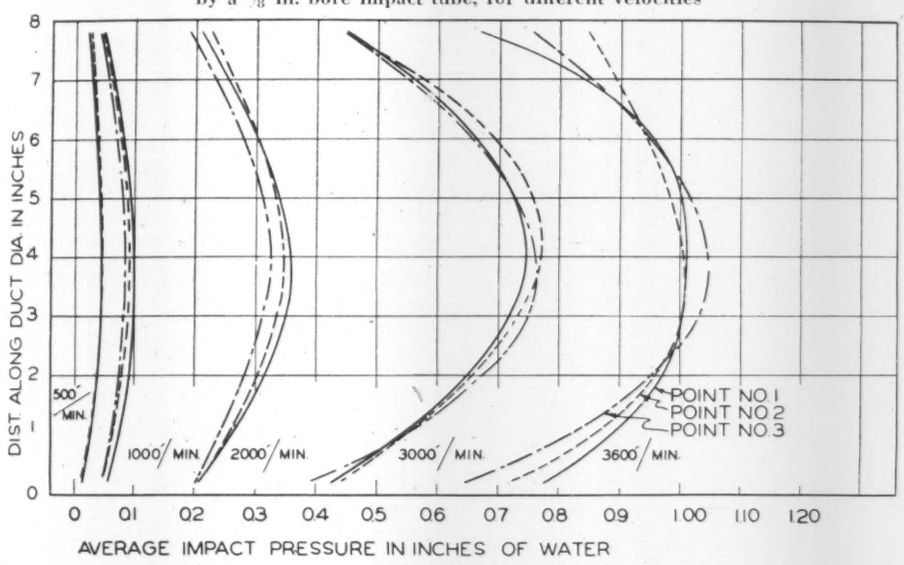


Fig. 12—Relation between position along the diameter of duct and impact pressure, as measured by an averaging impact tube at different velocities

and connected by a small brass tube. One cylinder is closed and is connected to the gage being calibrated. The other cylinder, open to the atmosphere, is fitted with an indicating pointer actuated by a micrometer screw. The micrometer screw was made for the manometer and is accurate to 0.00015 in. in 5 in. The micrometer head is so divided that readings are made to the nearest thousandth and estimated to the nearest ten-thousandth of an inch. Elimination of specific gravity determinations is obtained by using distilled water in the manometer.

The instrument is used by connecting one leg of the manometer to be calibrated to the closed cylinder of the micro-manometer. The other end of the manometer is open to the atmosphere as is the remaining cylinder of the micro-manometer, thus insuring a uniform reference pressure. When equilibrium has been established, the pointer is brought from above into contact with the water surface in the micro-manometer and a reading is taken. Pressure is then applied to the tube connecting the two manometers and again readings are taken. The difference in readings of the micro-manometer multiplied by two is the correct pressure in inches of water.

Indication of contact between the pointer and the water surface is given by a vacuum tube ammeter

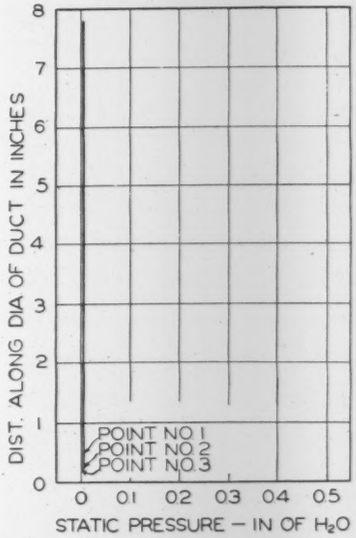


Fig. 13—Relation between position along the diameter of duct and pressure differential between a static tube inside the duct and the piezometer ring static pressure, at velocities from 3600 to 500 fpm

electric circuit. At the instant of contact the ammeter shows a large deflection. The instant of contact is indicated in this manner more nearly correctly than can be achieved by observing the pointer as it approaches the water surface.

## Appendix III

## Velocity Calculations for Nozzle on Test Set-Up

Notations:

$H$ = head, feet of air
$h$ = head, inches of water
$g$ = 32.2
$Q$ = quantity of air flowing
$d$ = density of air, lb/ft³
$V_n$ = velocity in nozzle, fpm
$V_d$ = velocity in duct, fpm
$A_n$ = area of nozzle, ft²
$A_d$ = area of duct, ft²
$D_n$ = diameter of nozzle
$D_d$ = diameter of duct
$C$ = nozzle coefficient = 0.99

General Nozzle Equation:

$$V = \sqrt{2gH}$$

$$V = \sqrt{2 \times 32.2 \times \frac{62.4}{d} \times \frac{h}{12}}$$

$$V = 18.3\sqrt{h/d}$$

$$V_n = 60\,CV = 1096.5\sqrt{h/d}$$

$$V_n = 1084.5\sqrt{h/d}$$

$$V_d = Q/A_d$$

$$V_d = V_n\,(A_n/A_d)$$

$$A_n/A_d = (D_n/D_d)^2$$

$$V_d = 1084.5\sqrt{h/d}\,(D_n/D_d)^2$$

## Appendix IV

Listed as follows are the dimensions of the shopmade splitters

which were tested in the miter elbow and the two-inch elbow.

FOR THE MITER ELBOW:

| VERTICAL HEIGHT | LENGTH | RADIUS |
|---|---|---|
| 7½ in. | 5½ in. | 2½ in. |
| 7½ in. | 4⅜ in. | 2½ in. |
| 5¼ in. | 4¼ in. | 2½ in. |
| 6 in. | 7 in. | 4 in. |
| 6⅜ in. | 5¾ in. | 3¼ in. |
| 7¼ in. | 6½ in. | 4 in. |
| 7⅛ in. | 7⅛ in. | 4 in. |
| 4⅜ in. | 2¾ in. | 1⅝ in. |

FOR THE STANDARD (TWO-INCH RADIUS) ELBOW:

| | | |
|---|---|---|
| 4⅜ in. | 2¾ in. | 1⅝ in. |
| 7 in. | 3 in. | 1¾ in. |
| 7 in. | 4½ in. | 3½ in. |
| 6¾ in. | 4¼ in. | 2½ in. |
| 7 in. | 7 in. | 4 in. |
| 7 in. | 5¾ in. | 3¼ in. |

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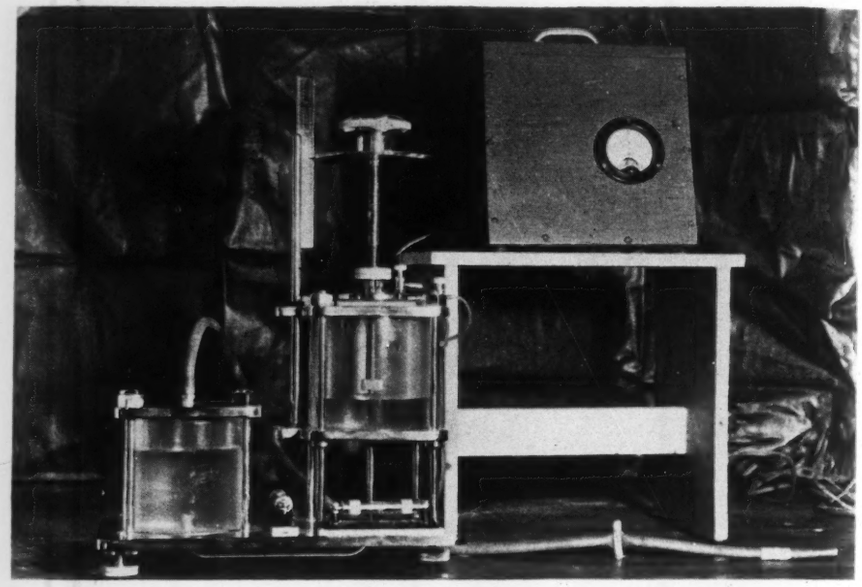

**Fig. 14—The Lehigh micro-manometer with electrical contact indication**

## *AN ENVIABLE RECORD*

L. A. Harding of Buffalo, N. Y., recently presented his resignation as Commissioner of the Department of Public Works of that city, after serving for three years and nine months. He leaves a record of accomplishments that will be difficult for his successors to equal and recently editorial tribute was paid to his efficiency as an administrator by the Buffalo newspapers.

*The Buffalo Evening News* said: "The resignation of Louis A. Harding as Commissioner of the Department of Public Works causes general regret. A highly capable engineer and an able executive, he has established a fine standard of efficiency in his department. He has been an exemplar of the spirit which regards public office as a public trust. . . These last three years and more the political tricksters have avoided the office of the Department of Public Works, for they have found the atmosphere there oppressive. Mr. Harding leaves the City Hall with the satisfaction of knowing that he enjoys the respect of all citizens who have a regard for good government."

*The Buffalo Courier Express* said editorially: "Public Works Commissioner Harding, who has submitted his resignation, has given the city an able and forthright administration of his department. He took office at a time when the city was in dire need of improved management. The wisdom of Mayor Holling's appointment has been amply demonstrated by the high quality of service rendered by the appointee. . . The Commissioner has his own reasons for retirement and by his record he has earned not only the right to quit public service, but also the public's gratitude for having kept the public's interest foremost during his tenure."

In a feature article by George Laird of the *Buffalo Evening News* staff he lists some of Mr. Harding's accomplishments in the following manner:

"Here are some of his achievements as remembered by the people who saw him daily—and he was one official who rarely was absent from his desk: When he started to prepare his first budget he was amazed at the number of employees who had jobs without work attached to them. City finances were at low ebb. With no fear of political retaliation, which made him respected throughout the City Government, he crossed upwards of 200 jobs —and the persons holding them—off the payroll. That "purge" saved thousands of dollars and interfered not at all with department efficiency. A few weeks later, contractors' plans for electrification of the Ward Pumping Station were brought to him for approval. An expert in that type of plant design, he glanced at the drawings, made a few penciled changes, eliminated several unnecessary items — and saved about $50,000."

Mr. Harding redesigned Niagara Street Bridge at a saving of $19,000; razed Elmwood Music Hall; designed and checked plans for the Airport Administration Building; supervised construction of the new Memorial Auditorium; completely motorized his department without a bond issue; with Budget Director John J. Egan, revised the street lighting contract at a saving of $100,000 a year; built part of Perry Boulevard after removing it from the talk stage; planned the Terrace Track Elimination Project to the point where it would soon get under way after 70 years discussion; made repairs on City Hall; and improved water supply to Park Lake.

"Commissioner Harding dropped all private business. He is returning to business now to write a book on the history of mathematics, part of which somehow he has managed to prepare during his busy four years."

# Comfort With Summer Air Conditioning

**Report of the ASHVE Research Technical Advisory Committee on Sensations of Comfort: Thomas Chester, *Chairman*; N. D. Adams, C. R. Bellamy, G. D. Fife, E. P. Heckel, Dr. W. J. McConnell, F. C. McIntosh, A. B. Newton, B. F. Raber and C. Tasker**

A GENERATION ago it was considered that ideal indoor atmospheric conditions should be similar to those of a perfect day in June. This was not very precise but the idea was good. Nature does provide such delightful days toward the end of Spring in the northern portion of the United States and earlier in the southern regions.

Human comfort is promoted in winter when the aerial envelope in contact with the skin below the clothing is easily maintained in a sub-tropical condition.

In summer the metabolic heat which can be lost by radiation and by convection is less, so it is necessary for the body to lose more heat by the evaporation of perspiration under natural atmospheric conditions. In hot and humid climates and also in densely-occupied enclosures in temperate climates in summer, the water vapor content of the atmosphere is too high for comfort. Under such conditions it is usually necessary to reduce both the dry-bulb and dew-point temperatures in order to assist the human body to dispose of heat by convection and by evaporation. It should also be noted that when the dry-bulb temperature is lowered, this results in a reduction of the average temperature of the interior surfaces of an enclosure and consequently more heat can be lost by the body by radiation.

It may be considered that the body's primary means of maintaining its required temperature is its regulated disposal of heat by radiation and by convection, and that heat loss by the evaporation of perspiration is a secondary method. It seems rational to assume the complete environment, atmospheric and physical, should be such that the body can freely use all its methods of heat disposal, and that it should not be compelled by environmental conditions to make undue use of one particular method, i. e., the evaporation of perspiration to the exclusion of the others.

An average adult in a state of little activity must lose heat at the rate of approximately 400 Btu per hour, both in summer and winter. Comfort is attained when this amount of heat can be lost without any sensation either of cold or heat.

In the early days of air conditioning for comfort in summer, the chief aim in most cases was to remove sensible and latent heat from the indoor atmosphere; usually too much of the former and too little of the latter, with the result that the air felt cold and noticeably damp. Even today individuals who are susceptible to colds and pulmonary troubles are advised by their physicians to avoid air conditioned theaters and restaurants.

During the past decade a great deal of research work has been carried on in order to ascertain indoor atmospheric conditions conducive to optimum comfort. Studies have been made over a considerable geographical range embracing Toronto, Minneapolis, Pittsburgh, New York, Washington, and San Antonio. The feelings and reactions to summer air conditioning of very many individuals have been carefully noted. The subjects included men and women, old and young, and attention was given to the clothing worn.

This resulted in the presentation to the Society of many papers containing a wealth of detailed information, with ably deduced summaries and conclusions, and disclosing a general conformity which amply warrants reliability.

During 1940 the Committee on Research requested the Technical Advisory Committee on Sensations of Comfort to review eleven papers and in consonance with them to make recommendations concerning indoor effective temperatures for summer occupancies. The papers given this consideration were:

1. ASHVE Research Report No. 1035—Comfort Standards for Summer Air Conditioning, by F. C. Houghten and Carl Gutberlet. (ASHVE Transactions, Vol. 42, 1936, p. 215.)
2. ASHVE Research Report No. 1055—Cooling Requirements for Summer Comfort Air Conditioning, by F. C. Houghten, F. E. Giesecke, C. Tasker and Carl Gutberlet. (ASHVE Transactions, Vol. 43, 1937, p. 145.)
3. Cooling Requirements for Summer Comfort Air Conditioning in Toronto, by C. Tasker. (ASHVE Transactions, Vol. 44, 1938, p. 549.)
4. ASHVE Research Report No. 1088—Summer Cooling Requirements of 275 Workers in an Air Conditioned Office, by A. B. Newton, F. C. Houghten, Carl Gutberlet and R. W. Qualley. (ASHVE Transactions, Vol. 44, 1938, p. 337.)
5. ASHVE Research Report No. 1101—Seasonal Variations in Effective Temperature Requirements, by F. E. Giesecke and W. H. Badgett. (ASHVE Transactions, Vol. 44, 1938, p. 559.)
6. ASHVE Research Report No. 1102—Shock Experiences of 275 Workers After Entering and Leaving Cooled and Air Conditioned Offices, by A. B. Newton, F. C. Houghten, Carl Gutberlet, R. W. Qualley and M. C. W. Tomlinson. (ASHVE Transactions, Vol. 44, 1938, p. 571.)
7. ASHVE Research Report No. 1103—General Reactions of 274 Office Workers to Summer Cooling and Air Conditioning by F. C. Houghten, A. B. Newton, R. W. Qualley and Edward Witkowski. (ASHVE Transactions, Vol. 44, 1938, p. 591.)
8. ASHVE Research Report No. 1127—Reactions of Office Workers to Air Conditioning in South Texas, by A. J. Rummel, F. E. Giesecke, W. H. Badgett and A. T. Moses. (ASHVE Transactions, Vol. 45, 1939, p. 459.)
9. ASHVE Research Report No. 1136. Summer Cooling Requirements in Washington, D. C., and Other Metropolitan Districts, by F. C. Houghten, Carl Gutberlet and Albert A. Rosenberg. (ASHVE Transactions, Vol. 45, 1939, p. 577.)
10. ASHVE Research Report No. 1153—Seasonal Variation in Reactions to Hot Atmospheres, by F. C. Houghten, A. A. Rosenberg and M. B. Ferderber. (ASHVE Transactions, Vol. 46, 1940.)

For presentation at the 48th Annual Meeting of the American Society of Heating and Ventilating Engineers, Philadelphia, Pa., January, 1942.

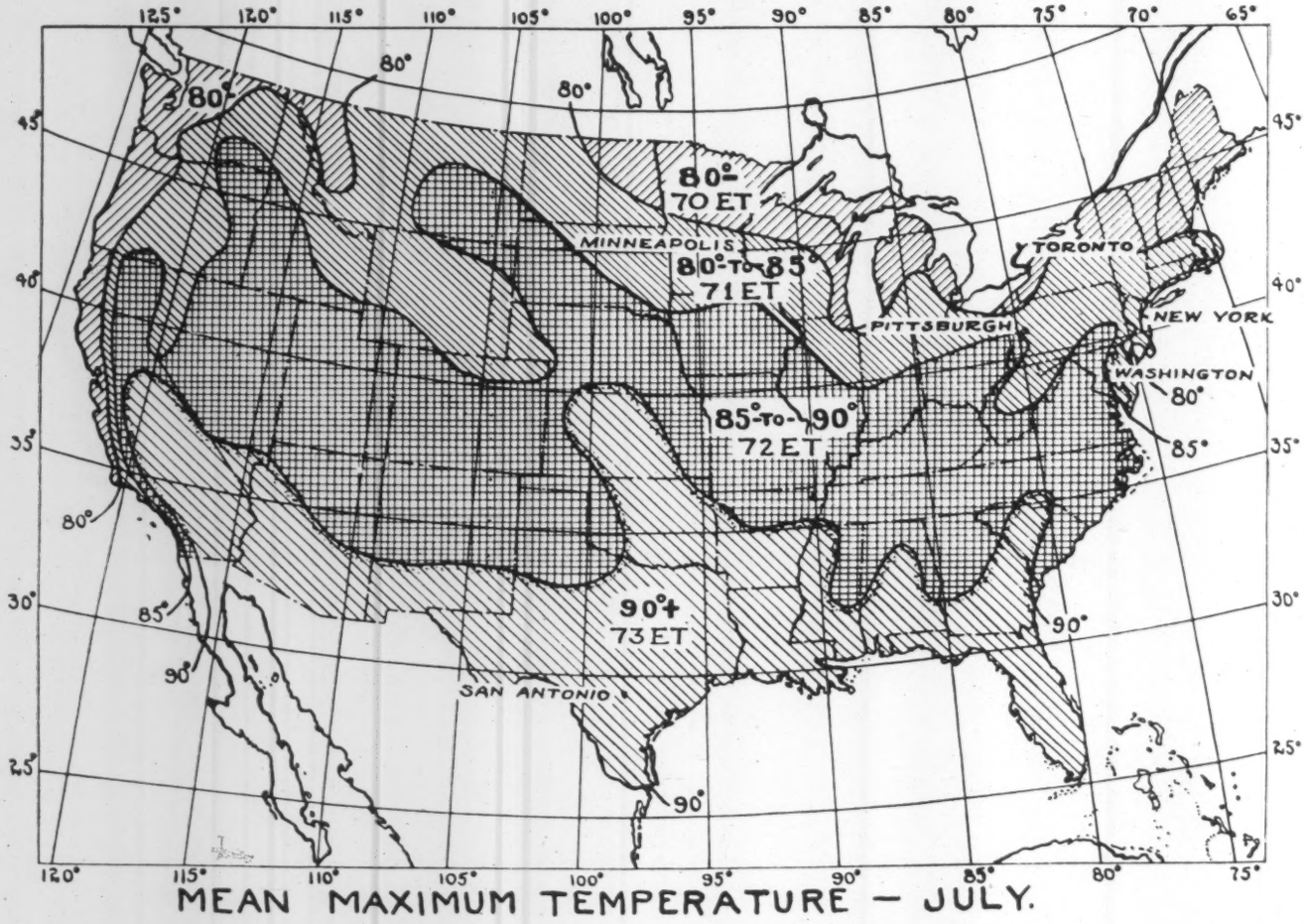


**Fig. 1—The above effective temperatures apply to buildings of common construction wherein no unusually large areas are affected by solar radiation. They are suited to long periods of occupancy by people in a state of very little activity or engaged in light sedentary or clerical work. Air velocities 15 to 25 fpm.**

11. Reactions of 745 Clerks to Summer Air Conditioning, by W. J. McConnell and M. Spiegelman. (ASHVE TRANSACTIONS, Vol. 46, 1940.)

Other papers of pertinent interest to this subject, which have been published recently by the Society, include:

1. ASHVE RESEARCH REPORT No. 1079—Physiologic Response of Man to Environmental Temperature, by F. K. Hick, R. W. Keeton and Nathaniel Glickman. (ASHVE TRANSACTIONS, Vol. 44, 1938, p. 145.)

2. Recent Advances in Physiological Knowledge and their Bearing on Ventilation Practice. Report of ASHVE Research Technical Advisory Committee on Physiological Reactions; C.-E. A. Winslow, *Chairman*. (ASHVE TRANSACTIONS, Vol. 45, 1939, p. 111.)

3. ASHVE RESEARCH REPORT No. 1111—Air Conditioning Requirements of an Operating Room and Recovery Ward, by F. C. Houghten and W. Leigh Cook, Jr. (ASHVE TRANSACTIONS, Vol. 45, 1939, p. 161.)

4. ASHVE RESEARCH REPORT No. 1151—The Peripheral Type of Circulatory Failure in Experimental Heat Exhaustion—The Role of Posture, by Robert W. Keeton, Ford K. Hick, Nathaniel Glickman and M. M. Montgomery. (ASHVE TRANSACTIONS, Vol. 46, 1940.)

5. ASHVE RESEARCH REPORT—Comfort Requirements for Low Humidity Air Conditioning, by F. C. Houghten, H. T. Olson and S. B. Gunst. (ASHVE JOURNAL SECTION, *Heating, Piping and Air Conditioning*, January 1941, p. 57.)

6. ASHVE RESEARCH REPORT—The Influence of Physiological Research on Comfort Requirements, by Robert W. Keeton, Ford K. Hick, Nathaniel Glickman and M. M. Montgomery. (ASHVE JOURNAL SECTION, *Heating, Piping and Air Conditioning*, March 1941, p. 188.)

Resulting from a review of the first eleven papers listed, it has been decided to recommend that the United States be divided into four zones from north to south, embracing a range of recommended indoor effective temperatures for design from 70 to 73 F. Fig. 1 shows the four zones and the corresponding recommended indoor effective temperatures.

Any idea that finality has been reached is expressly disclaimed, as there still remains a wide field for research. The recommendations apply to buildings approximately similar in design and construction to those in which the comfort investigations were made. In general these were of average modern type with normal walls and windows. Radical departures in design can involve variations in effective temperatures conducive to comfort.

Taking an extreme case, it is axiomatic that in an uninsulated galvanized iron structure a lower effective temperature would be desirable, in order to compensate for the effect of inward radiation from the boundary surfaces subjected to solar heat. Maintenance of the indoor effective temperature indicated for the geographical location of such a building would not be suitable, even with the solar heat gain properly computed in the required cooling load.

It will be obvious that with a rectangular building with long north

and south walls practically all glass, with short windowless east and west walls, and with a central partition wall running east to west, comfort conditions would be dissimilar in the two enclosures even if the same effective temperatures were maintained. The difference would be very noticeable with unprotected windows, and perceptible even if awnings were used, due to the difference in heat disposal by radiation from the occupants of the two rooms. In a paper[1] previously published by the Society, it is reported that one of the subjects stated in unscientific but unequivocal language:

> Air conditioning is okey, except along the south windows—when the sun shines in here it is hotter than outside. I suggest that you turn on more cold along the windows.

Theaters are windowless and usually have false ceilings with roof air spaces which afford protection against solar radiation. On the other hand, they contain many more persons per thousand square feet of floor area than office buildings. Closely packed occupants do not lose much heat laterally by radiation and loss by convection is less than in more sparsely occupied enclosures. Consequently, the selection of suitable effective temperatures for theaters offers scope for research.

Compensatory action for warm boundary surfaces can be produced by lowering the dry-bulb temperature while still maintaining the same effective temperature, but this of course involves increased relative humidity, and more satisfactory results can be obtained by lowering the effective temperature and maintaining a satisfactory relative humidity.

The recommended effective temperatures for the four zones shown by the map are for long occupancy periods. In places such as department stores and banks the comfort requirements of customers and employees differ. Individuals entering such air conditioned enclosures for short visits find an effective temperature best suited to the employees to be too cold for comfort. The chief cause of the cold shock upon entering is the quick cooling of perspiration laden clothing due to a hot and humid outdoor atmosphere. Evaporation tends to lower moist garments to the wet-bulb temperature of the conditioned air, which is considerably below the dry-bulb temperature.

For instance at 73 deg ET with 50 per cent relative humidity, the dry-bulb temperature is approximately 79 F and the wet-bulb temperature 66 F. It is not surprising that a pronounced chilling effect is produced but what can be done to counteract it is not definitely known. In cases where most persons enter an air conditioned enclosure at the same time, it would seem that initially the effective temperature could be about a mean between the outdoor effective temperature and that considered desirable for indoors, with gradual reduction to the required air condition in 20 to 30 min. Although experiments which have been made along this line have not been very successful, this need not dispose of the method and further investigation seems warranted. It must be admitted that in very many cases attempts to provide comfort in summer by means of air conditioning result in an initial feeling of pronounced discomfort.

As regards leaving shock it is generally considered that going into the hot and humid outdoor atmosphere from an air conditioned space is very unpleasant. Some engineers are of the opinion that this is due to the lag of the sweat glands in producing the necessary increased exudation of perspiration, and that to some extent this may result from the higher vapor pressure outdoors. Although evaporative cooling is sequentially the human body's secondary means of temperature control by heat disposal, it is much more potent than cooling by radiation and convection. The only method in sight at present for reducing leaving shock is gradual raising of the indoor effective temperature and this seems restricted to enclosures where all the occupants leave at about the same time. Maintenance of higher vapor pressures indoors is not feasible because this would involve higher dewpoints and increased relative humidities.

In the paper[2] previously mentioned, one individual is reported as having stated:

> After leaving the air conditioned office and entering the warm unconditioned places, perspiration began to flow freely, and for a short time a sort of dizzy feeling would be felt, but would soon wear off.

This was evidently an extreme case because in this same paper under the heading Discussion of Results the following appears:

> . . . Better than 35 per cent were frank in stating that they felt too warm for a while after leaving the air conditioned space. However, the duration of the hot reaction was in most cases not great, and no importance was placed on it. . . .

In another paper[3], it is stated:

> The warm shock that is felt by persons leaving an air conditioned building is a sensation that is not easily analyzed. It is needless to say that any condition is emphasized by contrast, but that should not necessarily mean that the condition is harmful because of the contrast . . . Practically the entire group expressed a feeling of discomfort immediately after leaving the building.

In general the investigations did not disclose that any harmful results ensued from leaving shock. From a practical standpoint it does not seem necessary for air conditioning engineers to be much concerned about this matter, as after all any discomfort felt upon leaving an air conditioned enclosure carries with it an appreciation of the comfort felt therein.

On the other hand entrance shock can be harmful to persons in delicate health and the public unquestionably reacts unfavorably to cold chills and feelings of clamminess, so research work in this direction seems worth while.

The GUIDE defines *Comfort Zone* (average) as the range of effective temperatures over which the majority of adults feel comfortable. Comfort does not suddenly diminish on each side of the optimum condition when the air contains either more or less total heat than is regarded as ideal. As shown by the various papers, comfort curves are parabolic, whether pertaining to the sensations of one individual under varied atmospheric conditions, or to the reactions of large groups of people to the same effective temperature. Ideal conditions are at the apices of such curves. Starting at

[1] ASHVE RESEARCH REPORT No. 1103—General Reactions of 274 Office Workers to Summer Cooling and Air Conditioning, by F. C. Houghten, A. B. Newton, R. W. Qualley and Edward Witkowski. (ASHVE TRANSACTIONS, Vol. 44, 1938, p. 591.)

[2] Loc. Cit. Note 1.

[3] ASHVE REPORT No. 1127—Reactions of Office Workers to Air Conditioning in South Texas, by A. J. Rummel, F. E. Giesecke, W. H. Badgett and A. T. Moses. (ASHVE TRANSACTIONS, Vol. 45, 1939, p. 459.)

the base of a curve on the cold side, seven primary and perceptible variations in comfort can be indicated graphically, as follows:

(1) Cold. 9 deg ET below optimum.
(2) Too cool. 6 deg ET below optimum.
(3) Comfortably cool. 3 deg ET below optimum.
(4) Ideal comfort. Optimum effective temperature.
(5) Comfortably warm. 3 deg ET above optimum.
(6) Too warm. 6 deg ET above optimum.
(7) Hot. 9 deg ET above optimum.

Individuals vary considerably in their opinions of the effective temperatures conducive to comfort. A great deal of information is given in the papers listed which it does not seem necessary to reprint in this digest.

It seems to have been proven that women prefer somewhat higher effective temperatures than men and that people over forty like higher temperatures than those under forty. This indicates the impossibility of establishing effective temperatures which are ideal for all people and all that can be done is to establish standards which will be objectionable to the smallest number. It is perhaps reasonable to attribute the difference in opinion of men and women in this matter to differences in clothing. It is rational to suspect that those men who feel somewhat too warm in an indoor atmosphere which suits the large majority of men and women are wearing heavier attire.

As previously stated the recommended effective temperatures apply to enclosures of common construction not unduly affected by solar radiation. They are suited to long periods of occupancy by people engaged in light sedentary work. Lower effective temperatures are necessary for persons engaged in occupations involving greater activity, as heat output increases with muscular effort.

The recommended effective temperatures also apply to enclosures in which the average air velocity is from 15 to 25 fpm. Perhaps in the future it will be possible to provide higher air velocities and thus go a little way in imitating nature, as the average air movement outdoors is in excess of 500 fpm. With present types of building construction it is not feasible in most cases to exceed the air velocities stated, without causing objectionable drafts.

The earliest recorded case of the employment of mechanical refrigeration in cooling air for comfort, was that of an ammonia system installed by a British firm at a Rajah's palace in India in 1887. In that country cooling by means of air movement has been in use from time immemorial. Manually operated punkahs or air beaters have generally been superseded by motor driven ceiling fans where electricity is available. In offices the overhead fans discharge air towards the ceiling and this provides good air movement without disturbance of papers on desks.

Desk fans and ceiling fans are still in wide use in the United States. In the drier regions their use along with evaporative cooling goes a long way in the alleviation of hot weather discomfort. Generous air velocities can be arranged as under such conditions drafts are not unpleasant, the effect being more or less similar to that of a breeze in the open air. Air conditioning by means of refrigerated air is in a different category, as currents of air introduced at 15 F or more below the maintained room temperature can be very objectionable.

In many tropical countries it is customary to provide buildings with wide verandahs to keep out the sun and rain, and to allow free air movement by means of large open windows. In air conditioning work the fascination of mechanical contrivances has almost obscured the old and inexpensive efficacy of removing heat from the skin by means of air circulation.

The question of relative humidity must receive consideration. The Comfort Chart developed in the Society's Laboratory and reproduced in THE GUIDE has stood very well the test of time. Recent investigations have disclosed its substantial accuracy for comparatively low relative humidities—in the neighborhood of 30 per cent. Although as a matter of scientific interest it was desirable to study comfort reactions with relative humidities up to 100 per cent, no one would think of using them for ordinary installations. The Comfort Chart shows a shaded area indicating a desirable relative humidity range of from 30 to 70 per cent. It is recommended that the relative humidity for summer comfort should not be in excess of 6 per cent and that more satisfactor results can be obtained with lowe relative humidities.

As regards to the four differen effective temperatures shown by th map, human adaptation to climatic conditions is generally accepted and it seems reasonable to believe that an individual who lives in the lowlands of the deep south where sugar cane is grown is more inured to heat and humidity than a person who lives near the Canadian border.

This is backed up by experiences in other parts of the world. It was found that summer comfort was obtained at much higher than 71 deg ET for foreigners residing in Shanghai, where climatic conditions are more severe as regards heat and humidity than those of the Gulf states in this country. The London School of Hygiene and Tropical Medicine recommends 85 F dry-bulb and 72 F wet-bulb for Nigeria, and this represents 78.2 deg ET, which is well outside the shaded area shown on the Comfort Chart.

For ships which travel the tropic oceans, a suitable condition of the outboard atmosphere assumed as a basis for design, which occasionally will be exceeded, is 90 F dry-bulb, 84 F wet-bulb, 80 per cent relative humidity, 82.5 F dew-point, and 11.95 grains of water vapor per cubic foot of moist air. For this condition the effective temperature is 86 deg. The high humidity is due to the fact that the ocean surface is at 82 to 85 F, with water temperatures in excess of these in ports and in narrow seas.

Under such climatic conditions it is suitable to maintain in the fully occupied air conditioned spaces, 82.5 F dry-bulb, 69 F wet-bulb, 50 per cent relative humidity, 62 F dew-point, and 5.96 grains of water vapor per cubic foot of conditioned air. The effective temperature for this condition is 76 F, which is well in excess of the standard recommended for the southernmost portion of the United States. Any appreciably lower effective temperature than stated would be too cold on the tropic oceans, because the passengers spend a great deal of time on deck in the open air and tend to become acclimated to its high total heat content.

The Comfort Chart shows an op-

timum of 66 deg ET for winter and 71 deg ET for summer. This is indicative of adaptation of the human body to climatic conditions, so to summarize the foregoing it seems rational to provide different optimum indoor effective temperatures in different portions of the United States.

In most places heat intensity is governed by the mid-day elevation of the sun above the horizon, so latitude is the chief determining factor. As regards altitude, the temperature of the atmosphere falls somewhat more than 5 F for each increase of 1,000 ft in elevation; the effect being modified by a slightly increased temperature of surfaces exposed to direct rays of the sun. The direction of prevailing winds also influences climates, as a wind is an atmospheric drift. In summer in this country the wind is mostly from the southwest. In the northern hemisphere sustained north winds result in lower air temperatures. In seaboard regions ocean currents also affect climates. On the west coast the current from the north tempers summer heat intensity, and on the east coast the Gulf Stream moderates the rigors of winter as far north as Cape Hatteras.

The endless procession of the seasons is due to the inclination of the earth's axis to the plane of its orbit around the sun. This causes the elevation of the sun at noon to be 47 deg higher on June 21 than on December 21 for regions in the northern hemisphere. Disregarding the reductions consequent to varying distances of penetration of the atmosphere, solar heat received per square foot of horizontal surface varies with the sine of the angle of the sun's elevation. It should be noted that places in high latitudes have more hours of sunlight per day in summer than places nearer the equator, and this introduces a variation factor. Maximum temperatures are reached in most regions about four weeks after maximum solar elevation, due to accumulation of heat in the atmosphere. Temperature drop during the night, due to radiation to interstellar space, is least when the sky is cloudy and greater when there are no clouds. Although entirely outside the field of air conditioning and in no way a substitute therefor, discomfort caused by summer heat can be somewhat alleviated by means of copious ventilation with relatively cool night air.

The effective temperatures shown in Fig. 1 are based on the design data given in the 1941 edition of THE GUIDE (p. 126). Design dry-bulb and wet-bulb temperatures are not maxima and consequently when peak summer weather conditions are experienced, the recommended indoor effective temperatures will be exceeded. Refrigerating plants are usually figured for design conditions and beyond their maximum capacity indoor dry-bulb and wet-bulb temperatures float with the total heat load. This is sometimes aggravated by decreased refrigerating effect in hot and humid weather, especially when atmospheric condensers are used.

With indoor effective temperatures as shown in Fig. 1, accurate thermal appraisals with recently available instruments, including globe thermometers, would disclose a greater difference between the northern and southern zones than is denoted by a difference of 3 deg in effective temperature. When it is considered that in one case the outside atmosphere is around 80 F and in the other case above 90 F it will be realized that room boundary internal surface temperatures will be somewhat more in excess of maintained indoor dry-bulb temperatures in the south than in the north. The differential which has a direct effect on comfort due to the lessened heat disposal from the occupants by radiation, is increased owing to the fact that the walls of buildings are usually thinner in the south than in the north.

As is well known, air conditioning is largely used in the Government buildings in Washington, D. C. Recent information concerning them, although not appearing in the papers reviewed, will be of interest.

For various outside dry-bulb temperatures the logs for July and August, 1940, indicate that the following effective temperatures were satisfactory for the Senate Chamber:

| OUTSIDE DEG F | INSIDE EFFECTIVE TEMP. |
|---|---|
| Under 76 | 70.5 |
| Between 76 and 80 | 71.0 |
| Between 81 and 85 | 71.4 |
| Between 86 and 90 | 71.8 |
| Between 91 and 95 | 72.7 |

The above effective temperatures are averages of eight readings for each group; each reading taken on a different day. The readings are scattered through the two months, with the exception of the last group which is confined to a period of about three weeks during the last of July. A study of the logs in sequence reveals a trend which is not apparent from the above table. This is, that for sudden changes in outside temperatures very little or no change in inside effective temperature is desirable. However, for a gradual increase in outside temperatures over a few days, it is desirable to increase the effective temperatures indoors.

This checks with the theory that the human body adapts itself to climatic conditions. After a pronounced heat wave in New York City which commenced July 19, 1940, and continued for 12 days, it was observed that when it abated somewhat, comfort was experienced in an un-air-conditioned room in which the atmosphere was at 85 F dry-bulb, 73 F wet-bulb, 56 per cent relative humidity, with interior surface temperature of the walls at 82 F. The equivalent effective temperature of 78.6 deg was conducive to comfort after tropical climatic conditions lasting 12 days.

Reverting to Washington, endeavor is made there to maintain dry-bulb temperatures of from 76 to 78 F with 50 per cent relative humidity throughout the buildings on Capitol Hill during the summer. With very few exceptions the room temperatures may be varied at will by the occupants by resetting the individual room thermostats. It has been found that people from southern states, older people, and women, require in general a higher temperature than others. From experience those in charge are strongly opposed to indiscriminate changing of indoor conditions to comply with any theoretical desirable relation between inside and outside temperatures, for the particular buildings under their supervision. It has been ascertained that too many variables are involved, such as the type of occupants, the length of occupancy, the rate of outdoor temperature change, etc. The logs for the Senate Chamber do not show the desirability of maintaining somewhat lower than normal effective temperatures to compensate for counter radiation between the occu-

pants. It is thought that this may be explained by the fact that the spacing of the occupants is much more generous than in an average theater, and also because the occupants are well above the average age. In the Capitol buildings the length of occupancy is from 3 to 9 hours, and it is considered important to maintain conditions desired by the regular occupants, without regard to the reactions of transients. It is held that the matter of entrance shock has been over-emphasized by others, as it is found of no importance. However, during the summer many of the entrance halls have higher temperatures than the interior rooms, and this is regarded as desirable. It is recognized that entrance shock is reduced by means of the warmer entrance halls.

In winter, maximum comfort is obtained by maintaining a 68 deg ET, which is 2 deg in excess of the optimum indicated by THE GUIDE, but it is thought that this again can be attributed to the fact that the occupants mainly considered are of more than average age.

This paper may be regarded somewhat as an interim report, as there is still a good deal to be learned concerning the interplay of radiation as affecting comfort in summer, between the occupants of air conditioned buildings and their physical environments. It is desirable that research be made in order to gain knowledge of important factors which at present are obscure or unknown. The Technical Advisory Committee on Radiation and Comfort has this subject under consideration and research work regarding it is now being carried on at the University of California.

In conclusion it must be stated that the four zonal effective temperatures shown in Fig. 1 should not be considered as inflexible requirements in all cases. Economic aspects have to be regarded and where limited financial resources prevent the installation of equipment capable of producing ideal comfort in summer, the aim should be to alleviate discomfort as far as funds permit.

## *BRITISH STANDARDS IN WARTIME*

Before the outbreak of war the *British Standards Institution*, which is the recognized center for the promulgation of all national British Standards, offered H. M. Government the services of the Institution, as a complete unit, in the national emergency. This offer was most cordially received and the various Departments of State were duly informed of the proposal.

At the outbreak of war the *BSI* realized that its peace-time procedure was inadequate to deal effectively with the demands imposed by the changed conditions, and especially by the need for rapid action. Therefore, a number of small executive committees were set up for the various sections of its work, these being made fully responsible for the preparation of any War Emergency Specifications the Institution might be called upon to undertake. The executive committees were given authority to restrict the usual wide consultation of industry to those interests directly concerned, and the reduction of the time usually given for comment on draft standards. It is, of course, understood that any British Standards issued under wartime procedure will come under review directly when peace comes again. As an indication of the value of the work of the *BSI*, it may be mentioned that the first issue of the War Emergency Specifications for tins and cans for food products and other commodities is estimated to have saved 40,000 tons of steel in the first year. The British Standards, specifically mentioned in the Government Order, have been issued with the approval of the Minister of Supply following the recommendations of the economy committees set up by the Materials Committee.

A further example of the employment of the *BSI* by the Ministry in effecting saving of material is the issue of a War Emergency Specification for bolts and nuts with smaller heads, which, it is estimated, will save many thousands of tons of steel a year.

The Institution is also rationalizing alloy and special steels and a committee under Dr. W. H. Hatfield, F. R. S., has drawn up a confidential report.

This war work has in no way prevented the *BSI* from maintaining its usual close relationship with the Dominions standardizing bodies, which have been kept in close touch with these developments.

## *J. J. NESBITT DIES AT 73*

John J. Nesbitt, president of John J. Nesbitt, Inc., Holmesburg, Pa., manufacturers of heating and ventilating equipment, died in Lankenau Hospital, Philadelphia, Pa., on September 11.

Mr. Nesbitt was born in Dublin, Ireland, in 1868 and received his early education there in private schools. He came to this country at the age of 15, and from 1885-1890 he attended Joseph School in Philadelphia, and entered the employ of Baker and Smith, New York, in 1890, where he remained for four years.

John J. Nesbitt

He was one of the leading contractors in the heating and ventilating of school buildings, and was in charge of the designing, erecting and managing of the central hot water plant in Atlantic City, and in more than 500 hotels, schools, office buildings, residences and power plants.

He devoted his entire business career to the field of heating, ventilating and air conditioning, and in 1917 he invented a schoolroom unit ventilator that met with wide acceptance.

Mr. Nesbitt joined the Society in 1923, and at the April 1938 meeting of the Council, action was taken to confer the grade of Life Membership upon him.

Mr. Nesbitt was a familiar figure at the Society meetings and will be missed by his many friends and associates in the heating and ventilating field.

The Officers and Council of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS extend their sincere sympathy to his sons, Albert J., John J., Jr., and Harry J. Nesbitt, and to his daughters, Mrs. John McElgin and Mrs. Richard Thurber, who survive.

# Recent Developments in Absorption Refrigeration

By Glen Miller*, Los Angeles, Calif.

TO many people the mention of absorption refrigeration calls to mind an antiquated and discarded form of the art of refrigeration. Ten years ago, this concept of absorption refrigeration might have been valid, but today this branch of the art is rapidly coming to the fore in many new and interesting guises. It is the purpose of this paper to present a brief review of the general absorption cycle, to bring out some of the underlying reasons for the revival of absorption refrigeration, to consider the types of equipment now being manufactured and to inquire briefly into the economic position of absorption refrigeration, both from an operating point of view and from the point of view of first costs.

In order to cover the points outlined for discussion, it is desirable to make certain definitions for the purpose of understanding this paper. In the same way that the *Coefficient of Performance* is used as the criterion of performance in compression refrigeration, so the energy ratio ($R$) can be used in a discussion of absorption refrigeration. Throughout this paper, ($R$) will be defined as the gross heat input (in the form of fuel) divided by the net refrigerating effect; for example, if a direct gas-fired absorption refrigeration machine takes 24,000 Btu per hour at the burner per ton of refrigeration, then ($R$) will equal 2.0. A two-fluid absorption refrigeration machine, for the purpose of this discussion, will be any machine in which there are employed only two working substances; one acting as a refrigerant and one as a solvent. A three-fluid absorption refrigeration machine will be defined as a machine in which there are two working fluids and one inert fluid. A self-energized absorption refrigeration machine will be defined as any absorption refrigeration machine in which no external source of power as electric motors or steam pumps, are used to cause the circulation of the liquids within the machine. A standard absorption refrigeration machine will be a two-fluid machine utilizing an electric motor for circulating the solution within the machine.

The most familiar absorption refrigeration machine is probably the ammonia-water machine which dates back to the early nineteenth century. In this machine the condenser, receiver and evaporator are ordinarily the same as they are in a compression refrigeration cycle, but the cold low pressure refrigerant vapor is absorbed into a water solution with a consequent liberation of heat in a device known as an absorber and thence pumped through a heat exchanger to the generator where, with the application of heat, ammonia gas is driven off the solution and is condensed in the condenser, after being rectified by one device or another. The hot weak liquor from the generator then passes back from the generator to the absorber through the heat exchanger and thus completes the cycle. The power required to pump the refrigerant vapor as dissolved in the liquor from the low pressure prevailing in the evaporator to the high pressure prevailing in the generator is but a small fraction of the power required to compress ammonia gas or vapor between these two pressures. A substantial amount of heat is required to liberate this gas from the liquid in the generator and consequently power is saved by the expenditure of substantial quantities of less expensive, lower grade energy in the form of heat.

The basic absorption refrigeration machine can be said to consist of an evaporator, an absorber, a heat exchanger, a generator and a condenser, in which at least two working fluids are used as refrigerant and solvent. The distinctive feature of the absorption cycle is that the low pressure refrigerant vapor is liquefied in the absorber and that refrigerant and solvent are separated in the generator to allow relatively pure refrigerant to pass to the condenser. The cycle is thermal rather than mechanical.

In the writer's opinion, there are two basic reasons for the newly aroused interest in absorption refrigeration machines and this interest has in turn caused manufacturers to expend large sums of money in the development of new and improved equipment. These two basic reasons are the desire of the fuel industries to acquire a summer load as an offset to the winter peak which exists in all of the fields of fuel utilization, and concern occasioned by the summer peak now developing on the electric power systems. This summer peak is caused in some parts of the country by air conditioning load and in other parts of the country by agricultural pumping load. In any case, the superimposing of the additional summer peak load of air conditioning is relatively unprofitable unless this peak can be compensated by winter usage, because the annual load factor for summer air conditioning equipment alone is on the order of 10 to 20 per cent. Fuel industries have, on the other hand, a winter peak with an annual load factor of from 15 to 30 per cent for heating and are in the position of being able to supply low grade energy for summer cooling without any extension in service facilities and consequently at a low energy cost. It is probable that in the next 10 years a condition will develop in which the average cost of power for air conditioning purposes will tend to rise because of the necessity for the power companies to recover the full cost of service to air conditioning consumers.

Many variations are possible on

*Member of ASHVE.
Presented at the Semi-Annual Meeting of the American Society of Heating and Ventilating Engineers, San Francisco, Calif., June, 1941.

the simple absorption cycle. In general, these variations take the form of:

(1) Different working fluids.

(2) The addition of other working fluids to the two used in a standard cycle, which will not be discussed.

(3) The addition of inert fluids.

(4) The elimination of the pump required to circulate the solution in the standard cycle.

(5) Improvement in design and construction of the standard ammonia-water machine.

(6) Reabsorption cycles which will not be discussed here.

The most interesting recent developments in the absorption refrigeration field are probably the development of the three-fluid refrigeration cycle, the improved ammonia-water machine, the methylene chloride machine, the water-lithium chloride machines, the many self-energized ammonia-water machines, and the developments by several manufacturers in the use of halogenated hydro-carbon compounds as the working fluids in absorption refrigeration machines.

The most interesting three-fluid unit uses ammonia as refrigerant and water as a solvent with a third inert fluid to balance the total pressure in the high and low side of the system. This unit is the familiar gas-fired domestic refrigerator. The largest units of this type in use commercially are of one-half ton capacity.

Refinements have been introduced in the two-fluid ammonia-water machines such as the substitution of a bubble plate column for the conventional generator - rectifier system.

In the past 10 or 12 years, much work has been done on the substitution of other working fluids for the ammonia-water used in the old absorption machine. The reports of Zellhoefer[1] cover this work well and thoroughly. The refrigerant used by one firm is methylene chloride and the solvent is dimethyl ether of tetraethylene glycol. Some properties are shown in tabulation and operating data for one season's operation on a nominal fourteen ton machine are shown in Table 1.

[1]Summer and Winter Air Conditioning by Low Pressure Steam, by G. F. Zellhoefer. (ASHVE Journal Section, *Heating, Piping and Air Conditioning*, April, 1937, p. 265.)

For normal air conditioning service:

Low side pressure.....
.....22 in. to 24 in. hg vacuum

High side pressure.....
.....About ½ lb per square inch gage

Approximate heat balance (Btu per ton per minute)

| | *Heat into cycle* | *Heat out of cycle* |
|---|---|---|
| Evaporator ..... | 200 | |
| Absorber ..... | | 330 |
| Pump ..... | 2 | |
| Generator ..... | 365 | |
| Condenser ..... | | 237 |
| | 567 | 567 |
| Heat transferred in exchanger .. | 290 | 290 |

Among the most efficient absorption machines which have yet been produced are the water - lithium chloride self - energized machines. Water is the refrigerant and lithium chloride is the solvent. Circulation of the liquors is produced by a jet pump or percolator. High efficiencies ($R = 2$ or less) are obtained and the cooling can be applied directly to an air stream or a secondary refrigerant can be used. The cycle offers some possibilities for greatly reducing the condensing water requirements by increasing the temperature (and pressure) of condensing, but this development is still in the experimental stage. One such unit was field tested extensively last year and is in limited production this year.

A test of one field installation at Montebello, Calif., gave the following results:

| | |
|---|---|
| Air Flow ..... | 2,160 Cfm |
| Air to evaporator..... | 74 F Dry-Bulb |
| | 61 F Wet-Bulb |
| Air from evaporator..... | 56 F Dry Bulb |
| | 53 F Wet-Bulb |
| Condensing water | |
| Flow rate ..... | 89 lb per minute |
| Temperature in ..... | 71 F |
| Temperature out ..... | 95 F |
| Gas consumption ..... | 83.4 cu ft per hour |
| | 104,000 Btu per hour |
| Net refrigeration ..... | 838 Btu per minute |
| | 4.19 tons |
| Energy ratio ..... | $R = 2.07$ |

A 240-hour sustained test showed an average value for ($R$) of 1.94 including all losses and the results were extrapolated for a full season of 1,000 hours.

| | |
|---|---|
| Total cooling period (Equivalent full hours)..... | 1,000 |
| Total refrigerating effect (In ton hours)..... | 4,470 |
| Total gas consumption..... | 101.9 Mcf |
| Total water consumption..... | 118.7 Mcf |
| Power consumption (Including fan operation)..... | 1,380 kwhr |

The 240-hour sustained test figures were obtained by an artificial loading by the use of unvented space heaters. In unvented space heaters burning natural gas, the latent heat-total heat ratio is approximately 0.11.

Many attempts have been made to adapt the ammonia-water system to a small hermetically sealed refrigeration system without resorting to inert fluids covered by patents but none have been marketable. Intermittent operation and cyclic pressure balancing by weak solution inflow to a balancing chamber are among the methods used.

At this time, work is being carried out on the development of absorption machines using as both refrigerant and solvent halogenated hydro-carbon compounds. Some of these systems appear to be promising. Such combinations of chemicals lend themselves to either standard machines or to self-energized machines and some information is given on them as on other machines in Table 2.

Considering the economic problems which rise out of variations between absorption and compression equipment in first cost, the problem is complicated and presumably must be studied in each particular location for each particular machine. In general, it can be said that the condenser and evaporator are the same on both the absorption and compression machines. In general, in the water cooling mechanism, whether it be well water or city

**Table 1—Summary of One Year's Operation**

| | |
|---|---|
| Machine size..... | 14 ton |
| Steam supply..... | Gas fired boiler |
| Volume cooled..... | 43,400 cu ft |
| Gas Consumption | |
| Cooling..... | 375.1 Mcf |
| Heating..... | 306.2 Mcf |
| Electric Consumption[a] | |
| Cooling..... | 6,792 kwhr |
| Heating..... | 4,454 kwhr |
| Cooling | |
| Gas—1000 cu ft vol.. | 8.64 Mcf per year |
| Elec.—1000 cu ft vol.. | 156.5 kwhr per year |

[a]Includes fan power and evaporative condenser power.

This installation at Santa Ana, Cal., is in a territory where the normal design conditions are: Outside 95 F dry-bulb and 70 F or 71 F wet-bulb; Inside 80 F dry-bulb and 65 F wet-bulb for cooling. For heating, normal design temperature is 30 F. The climate is such that both heating and cooling will be required on at least 50 days of the year. Automatic control switches the machine from heating to cooling and back to heating with the thermostat settings as follows: (1) Below 75 F heating; (2) 75 F to 79F ventilation; (3) Above 79 F cooling.

Approximately 900 equivalent full hours of operation are the normal heating load and ordinarily the cooling operation is considered to be 800 to 1000 equivalent full hours of operation, both on an annual basis.

water wasted to the sewer, spray pond, atmospheric tower, or induced or forced draft tower, the heat removal will be from 1.7 to 3.0 times as great on absorption machines as on compression machines, considering only those machines that have energy ratios such as to make them an economic factor in the refrigeration field. An absorption machine with an energy ratio of 1.43 will have to dissipate in the cooling water at least 1.7 times as much heat as a compression machine, while an energy ratio of 3.43 will require the dissipation of three times as much heat in the cooling water as in the compression machine, assuming a boiler efficiency of 80 per cent. The cost of heat exchanger, boiler, absorber and pump ordinarily must be equated against the cost, in the standard compression cycle, of the compressor. In many cases, the boiler will already be in existence, or steam for driving the absorption machine may be available from district steam mains and consequently the cost of the boilers frequently does not enter the computation. It is probable that in the small and medium sizes (up to 25 tons capacity) the absorption equipment can be produced at or below the level of present costs of the standard compression cycle. The major differential in cost then probably will be in the water disposal means and this will add, on the basis of twice the water requirements for the absorption machine, about $20.00 per ton of installed capacity.

Operating costs of absorption equipment as contrasted with compression equipment are influenced by several factors, the most important of which are power costs, fuel or steam costs, water costs and the value ($R$) for the particular machine. For purposes of comparison, it can be assumed that a compressor will require one kilowatt hour per ton of refrigeration and the condensing water auxiliaries will require approximately 0.1 of a kilowatt hour per ton hour. The water make-up costs are more difficult to figure, but for the sake of analysis, one can assume that 1 cu ft of water at 10c per Ccf will be required for make-up per ton per hour on compression refrigeration. For the absorption refrigeration machine the power to circulate the liquor will be on the order of 0.1 of a kilowatt per ton and the auxiliary cooling water power will vary with the value of ($R$) as will the make-up water.

With these assumptions the equation for equal operating costs (neglecting for the moment fixed charges and maintenance) is as follows:

Let $A$ = Average power cost in cents per kwhr

Let $B$ = Average fuel cost in cents per therm

Let $R$ = Energy ratio as previously defined

*Note:* In most instances, the average rate for the power required will be approximately equal for both compression and absorption cycles and will be so considered in the following equation for equal operating costs.

Then, for equal operating costs, on a standard cycle—

$$1A + 0.1A + 0.1 = \frac{12{,}000R}{100{,}000} B + 0.1A + R.1A + R.1$$

This equation can be reduced to—

$$A = \left(\frac{RB + 0.85R - 0.85}{8.5}\right)\left(\frac{10}{10\text{-}R}\right)$$

*Note:* The additional power for circulation of cooling water and the additional cooling water make-up required will vary not directly with $R$ but in accordance with the factor $\frac{(1 + 8/10R)}{1.25}$. For the purpose of this comparison, however, the use of the proportionality factor $R$ is sufficiently accurate.

This relationship is shown on Fig. 1 for varying values of $R$.

For a self-energized machine, the equation is:

$$A = \left(\frac{RB + 0.85R - 0.85}{8.5}\right)\left(\frac{10}{11\text{-}R}\right)$$

This relationship is shown on Fig. 2.

No consideration of the economic picture of refrigeration equipment is complete without some consideration of the problem of maintenance. It has been the author's observation that on the self-energized hermetically sealed absorption machines that maintenance may easily be sub-

**Table 2—Information on Several Commercial and Experimental Machines**

| Make of Machine | Refrigerant | Solvent | Net Btu Input per Ton | Pumping Hp for Solutions | Pumping Hp for Cooling Water (Based on Atmospheric Tower) | Energy Ratio | Equivalent Electric Cost for Fuel at 4c per Therm | Tonnage Range |
|---|---|---|---|---|---|---|---|---|
| A | Methylene Chloride | Dimethyl Ether of Tetraethylene Glycol | 18 to 20M | 0.1 | 0.2 | 2.1 | 1.4c per KWHR | 5 to 40 |
| B | Water | Lithium Chloride | 16M | 0 | 0.15 | 1.67 | 1.0c per KWHR | 3 to 5 |
| C | Ammonia | Water | Approx. 30M | 0 | Approx. 0.33[a] | 3.7 | 2.8c per KWHR | About 2.5 |
| D | Ammonia | Water | 21M | 0.1 | 0.16 | 2.3 | 1.6c per KWHR | 100 to 1,000 |
| E | Ammonia | Water | 45M | 0 | [b] | 5.0 | 4.6c per KWHR | 0.1 to 0.5 |
| F[c] | Sulphur Dioxide | Acetone | Approx. 13.7M | — | — | 1.43 | 0.7c per KWHR | Experimental |
| G[c] | Freon-114 | Carbon Tetrachloride | 28.8M | — | — | 3.0 | 2.3c per KWHR | Experimental |
| H | Freon-21 | Tetrachlor-ethylene | — | — | — | 2.5 | 1.8c per KWHR | Experimental |

[a] Forced draft tower.
[b] Water is ordinarily wasted with this unit.
[c] This combination of working fluids probably cannot be used commercially because of corrosion and polymerization problems.

A. Data represent the average of manufacturer's ratings and have been checked by the author to very close tolerances in the field. There have been installed about 3500 tons capacity of this equipment.
B. Data represent manufacturer's rating and have been checked to within the limits of error of a field test by the author. The manufacturer of this equipment reports that about 1500 tons capacity will be marketed this year, most of which will be used in residential and small commercial comfort cooling installations.
C. Data obtained on a field test of an experimental machine by the author.
D. Data are manufacturer's rating on large machines of which 8000 tons refrigerating capacity have been sold in recent years.
E. Data represent the results of field tests on a unit offered for sale in the commercial refrigeration field.
The figures given for F, G and H are calculated figures and at this time no adequate test data are available. In the case of F and H, laboratory tests have been run but the data are not reliable.
Note: Except for D and E the data refer to air conditioning conditions: $T_e$ = 35F, $T_c$ = 90F. For D and E $T_e$ somewhat lower.

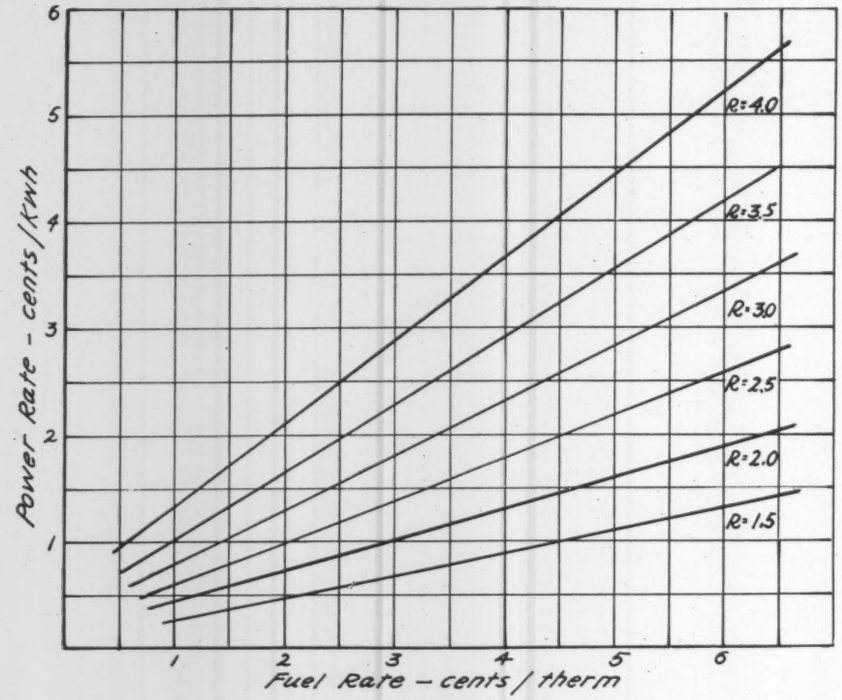


Fig. 1—Shows rates for equal operating costs compression vs. absorption refrigeration

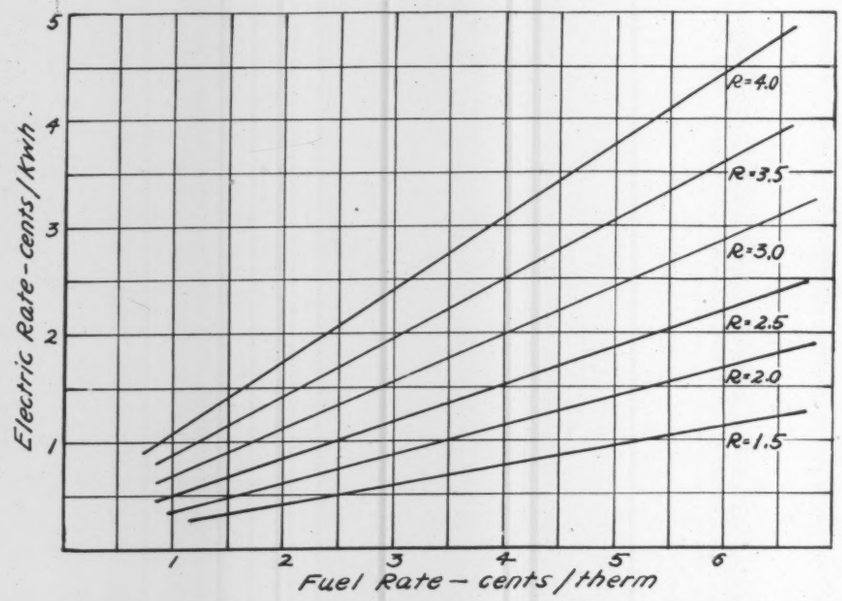


Fig. 2—Shows rates for equal operating costs compression vs. self-energized absorption machines

stantially less than on average compression equipment. With high vacuum, open type, standard absorption machines utilizing refrigerant-solvent combinations other than ammonia-water, maintenance has sometimes presented a very difficult problem. On the other hand, refrigerant loss in such systems is almost unheard of. As a general rule, it can be considered that maintenance will be no greater in the absorption machine than in the compression machine of similar capacity.

The complete economic problem is one which must be analyzed for each particular machine in each particular location, but it is hoped that the figures and observations presented may be helpful.

## Conclusions

In conclusion, it can probably be definitely stated that—

1. The new interest and the recent developments in absorption refrigeration have been brought about by the economic pressure of unused fuel capacity in the summer time, and the economic pressure of a new summer peak in electric power send-out.

2. The possibilities for development of new absorption cycles are very great and it can be expected that considerable research and development will be done in this field.

3. Absorption refrigeration cycles will be very prominent in the long awaited penetration of the residential field by air conditioning equipment because, at least at the present time, it is in this field that the new absorption machines have made the greatest developments and the best economic showing as will be seen by the table and charts appended.

4. Absorption refrigeration is once again an economic factor in the refrigeration field since within six years about 12,000 tons of capacity have been installed to serve such varied functions as oil refining and residential cooling.

To

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# Application of Gas Fuel to Heating Boilers

By R. L. Grutzmacher,* San Francisco, Calif.

THE use of gas fuel for heating boilers is not new. The problems of burner design and application have been discussed in other reports of this Society[1]. For purposes of this discussion, the developments of burners may be summarized to emphasize the relationship of burner design and arrangement to automatic control. The vertically fired gas burner units, as shown in Fig. 1, is now much used for firing gas fuel in heating boilers.

The most important feature of the vertically fired unit is its ability to effect rapid complete combustion in a relatively limited furnace volume with a minimum amount of excess air. Fundamentally this is the primary function of any burner and it may be explained in terms of the process of combustion.

Combustion is simply the chemical combination of fuel with oxygen in a reaction which liberates heat. The oxygen required for this reaction is obtained from the air, so that if a fuel is being supplied to a furnace at a given rate, air must also be supplied at a proportional rate. A unit of fuel containing carbon perfectly burned will produce a definite unit of $CO_2$. This unit is expressed as a percentage by volume of $CO_2$ of the flue gases and it is an indication of the maximum heat developed.

It follows that losses will occur if an excess amount of air is supplied even though combustion is complete. These losses are shown graphically in Fig. 2. Conversely, due to the law of combining weights, any fuel in excess of that for which the necessary air is supplied will go through the furnace unburned and its total heat value

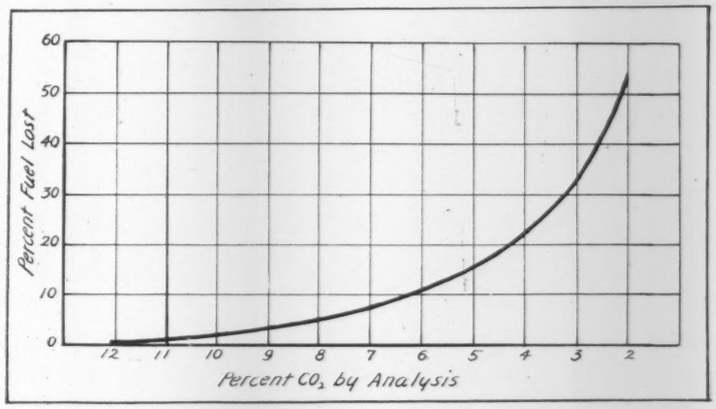



Fig. 2—Graph showing fuel losses due to excess air

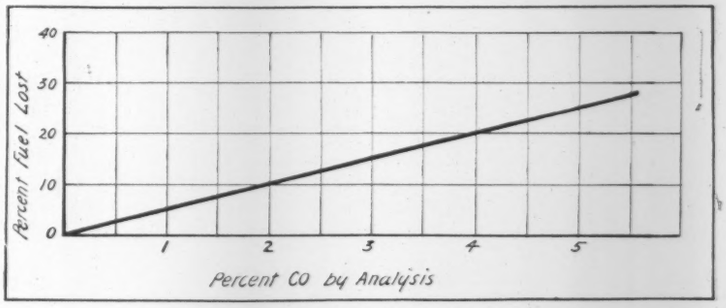



Fig. 3—Graph showing fuel losses due to CO

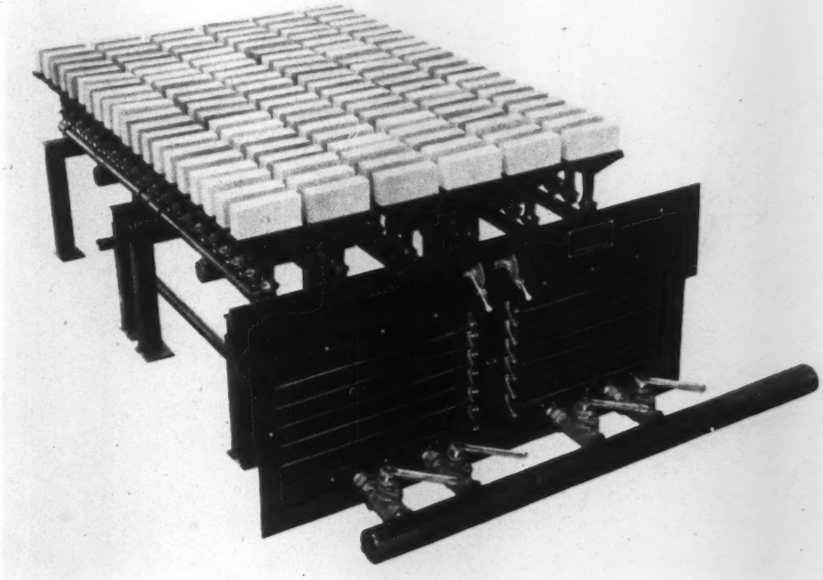


Fig. 1—Photograph of vertically fired gas burner

*Natural Gas Equipment, Inc., San Francisco, Calif.

[1]Vertical Conversion Gas Burners, by L. S. Regan. (*Heating, Piping and Air Conditioning*, Vol. 12, June, 1940, p. 362.)

Presented at the Semi-Annual Meeting of the American Society of Heating and Ventilating Engineers, San Francisco, Calif., June, 1941.

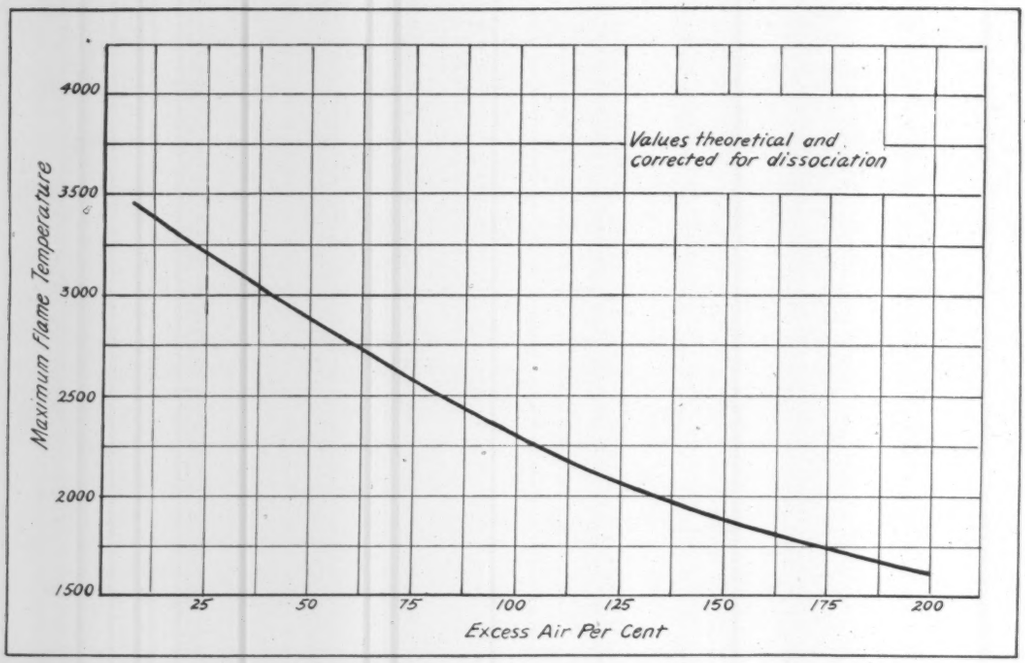



Fig. 4—Flame temperature chart

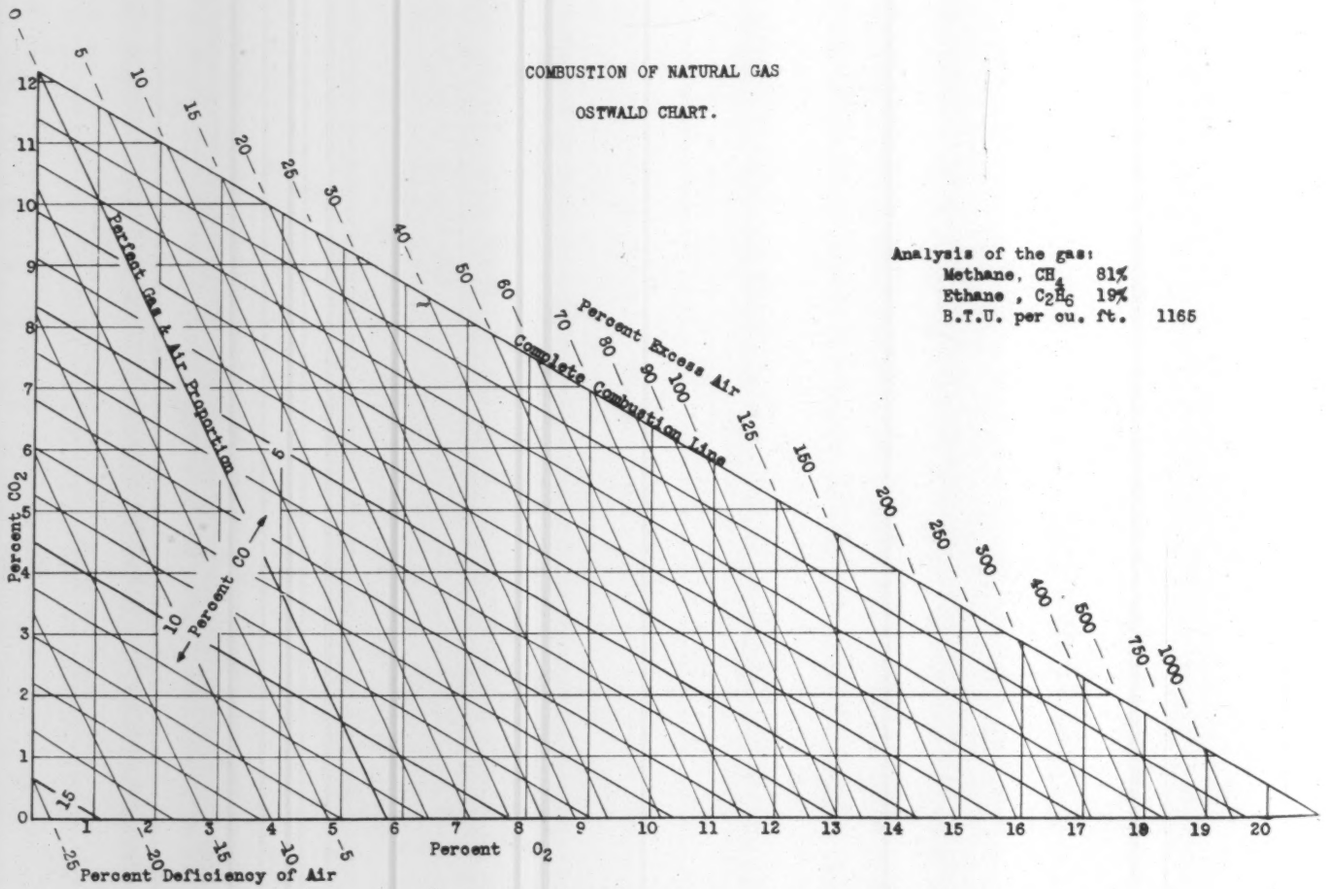



Fig. 5—Chart used for checking combustion analysis

**Table 1—Thermal Efficiencies Obtained Using Intermittent Control**

| Installation | 1 | 2 | 3 | 4 | 5 | 6 | 7 |
|---|---|---|---|---|---|---|---|
| Boiler Type | Cast-Iron | Cast-Iron | Steel | Steel | Steel | Steel | Scotch |
| Boiler Rtg. Sq Ft Rad. | 1815 | 3600 | 10000 | 6080 | 5280 | 1800 | 66 hp |
| Burner Type | V.V.[a] | M.J.V.[b] | M.J.V. | V.V. | V.V. | V.V. | V. Horiz.[c] |
| Gas Pressure | 8.2" WC | 4 oz | 8 oz | 3.7" WC | 3.1" WC | 4.1 | 1.5 lb |
| Gas Input Cu Ft-Hr | 545 | 575 | 2890 | 1500 | 1050 | 450 | 2500 |
| Per cent Rating | 100 | 57 | 100 | 82.5 | 76.6 | 96 | 94 |
| $CO_2$, Per Cent | 10.2 | 9.6 | 11.0 | 9.5 | 10.0 | 9.6 | 9.5 |
| $O_2$, Per Cent | 3.3 | 4.2 | 2.0 | 4.2 | 3.8 | 4.2 | 4.5 |
| CO, Per Cent | 0.0 | 0.0 | 0.0 | 0.0 | 0.0 | 0.0 | 0.0 |
| Excess Air, Per Cent | 17. | 24.5 | 10.0 | 25.0 | 19.5 | 23.0 | 25.0 |
| Stack Temp. F | 575 | 555 | 480 | 470 | 395 | 450 | 480 |
| Stack Draft In. WC | 0.04 | 0.05 | 0.035 | .02 | 0.03 | 0.01 | 0.12 |
| Efficiency, * | 78.5 | 78.4 | 81.4 | 80.5 | 82 | 81 | 80.2 |
| Control Type | Inter | Inter | Inter | Inter | Inter | Inter | Inter |
| Control On A.M. | 4:00 | 6:00 | ..... | 4:30 | 6:30 | 5:30 | ..... |
| Off A.M. | 9:00 | 9:00 | ..... | 9:00 | 10:30 | 9:45 | ..... |
| On P.M. | 4:00 | 4:00 | ..... | 4:00 | 3:30 | 4:30 | ..... |
| Off P.M. | 10:00 | 9:00 | ..... | 10:00 | 10:30 | 8:45 | ..... |

* Efficiency $= 100 \times \frac{\text{Heat input} - \text{Stack loss}}{\text{Heat input}}$

[a] Venturi Vertical Type.
[b] Multiple-jet Vertical Type.
[c] Venturi Horizontal Type.

will be lost. Fig. 3 is the curve indicating percentage of fuel loss when unburned gases are present.

It will be noted from the curves in Figs. 2 and 3 that the losses due to unburned fuel are correspondingly greater than the losses due to excess air. However, the factor of excess air is of more interest because gas analyses less frequently show unburned fuel. Excess air not only directly increases the stack losses by increase of the sensible heat of the gases, but also indirectly by decreasing the flame temperatures. Fig. 4 indicates the effect of excess air on flame temperature of an average natural gas. The successful application of gas fuel in vertically fired units is largely dependent upon effective heat transmission by radiation from the flame to the direct heating surfaces of the boiler, which quantity is directly proportional to the differences in absolute temperatures to the fourth power. For this reason, if complete combustion is obtained the excess air should not exceed from 20 to 25 per cent. A useful means of checking combustion analysis is the chart shown in Fig. 5.

The arrangement of the burners with reference to the heating surfaces of the boilers is greatly improved through the use of vertically fired units. The accomplishment of rapid complete combustion effects a short flame length, and the burners may be suitably placed in position with respect to the water legs and crown sheet to afford more effective and longer travel of the products of combustion. Burner arrangements will depend upon the burner characteristics and boiler design. Generally the manufacturer of such equipment has data or recommendations for the most effective use of his equipment.

Automatic control of gas fired equipment has engaged the attention of all gas engineers and is discussed herewith. The measure of adaptability of any fuel, or the equipment employed to utilize it, is the relative facility with which it may be efficiently controlled.

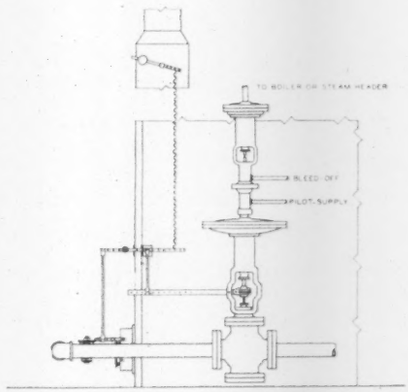


**Fig. 6—Illustration of a typical modulating control**

The question of relative efficiency of intermittent (full off and full on) and modulating control arises frequently. Each of these types has definite spheres of application. Installations which are intermittent in operation or have intermittent load demands, will almost invariably operate more efficiently, as judged from fuel usage, if they are intermittently controlled. Table 1 is a tabulation of test data taken from a large number of tests, which indicates the average high efficiencies obtainable with intermittent control. Hundreds of installations of this type provide the conclusion that regardless of boiler type or size, intermittent controls of vertically fired units are more efficient than modulating controls. The only limitation to this assertion is that the boilers are sized to the load and the burner capacity is within limits to afford safe and proper ignition. The maximum size or capacity of the burner is limited only by this latter qualification. A further conclusion reached by the study of these results is that vertical multiple jet or vertical venturi type burners are equally efficient when they have been properly applied.

There are instances in which load requirements may cause short maximum peak periods with normal operating periods and frequent off periods. To meet these conditions requires either step control or intermittent sectional control to avoid rapid cycling or short firing intervals. This is essentially the operation of two burner sections, each operated independently of the other. Table 2 gives the test data on two installations controlled by the intermittent sectional method. Each burner bank is individually controlled. It will be noted that in

**Table 2—Thermal Efficiencies Obtained Using Sectional Intermittent Control**

| | 1 | | | 2 | | |
|---|---|---|---|---|---|---|
| | A | B | C | A | B | C* |
| Boiler Type | Steel Firebox | | | Steel Firebox | | |
| Boiler Rating | 17420 Sq Ft Rad. | | | 80 Hp | | |
| Burner Type | Vertical Multiple-jet | | | Vertical Multiple-jet | | |
| Gas Input Cu Ft/Hr | 4100 | 2650 | 1900 | 4750 | 3400 | 2350 |
| $CO_2$, Per Cent | 11.0 | 7.5 | 6.0 | 9.5 | 9.0 | 8.0 |
| $O_2$, Per Cent | 1.9 | 7.8 | 10.5 | 4.3 | 5.3 | 7.1 |
| CO, Per Cent | 0.0 | 0.0 | 0.0 | 0.0 | 0.0 | 0.0 |
| Excess Air, Per Cent | 9.0 | 54.0 | 92.0 | 24.0 | 32.0 | 47.0 |
| Stack Temp. F | 330 | 305 | 295 | 650 | 580 | 475 |
| Stack Draft In. WC | 0.08 | 0.08 | 0.08 | 0.11 | 0.13 | 0.03 |
| Efficiency, Per Cent | 84.8 | 83.4 | 82.2 | 75.9 | 77.1 | 78.8 |
| Percent Boiler Rtg | 82 | 53 | 38 | 152 | 110 | 75 |

Tests A: Both Sections firing.
Tests B: Large Sections firing.
Tests C: Small Sections firing.
Large section approximately 2/3 capacity.
Small section approximately 1/3 capacity.

Table 3—Thermal Efficiencies Obtained Using Modulating Control

| | | | |
|---|---|---|---|
| Boiler Class | Steel Firebox | | |
| Boiler Rating | 18000 Sq Ft Rad. Max. | | |
| Burners | Vertical Venturi Type | | |
| | Test A | Test B | Test C |
| Gas Pressure Oz | 14 | 9.5 | 4 |
| Gas Input Cu Ft/Hr | 6000 | 5150 | 3550 |
| $CO_2$, Per Cent | 9.0 | 9.4 | 9.6 |
| $O_2$, Per Cent | 5.8 | 4.0 | 4.0 |
| CO, Per Cent | 0.0 | Tr | 0.0 |
| Excess Air Per Cent | 33 | 20 | 21 |
| Stack Temp. F. | 540 | 510 | 430 |
| Stack Draft In. WC | 0.25 | 0.18 | 0.07 |
| °Efficiency Per Cent | 78.1 | 78.1 | 81.5 |
| Per cent Boiler Rtg | 124 | 106 | 72 |

Test A, Maximum Fire.
Test B, Normal Operating Range.
Test C, Holding Fire.

$$\text{Efficiency} = 100 \times \frac{\text{Heat input—Stack loss}}{\text{Heat input}}$$

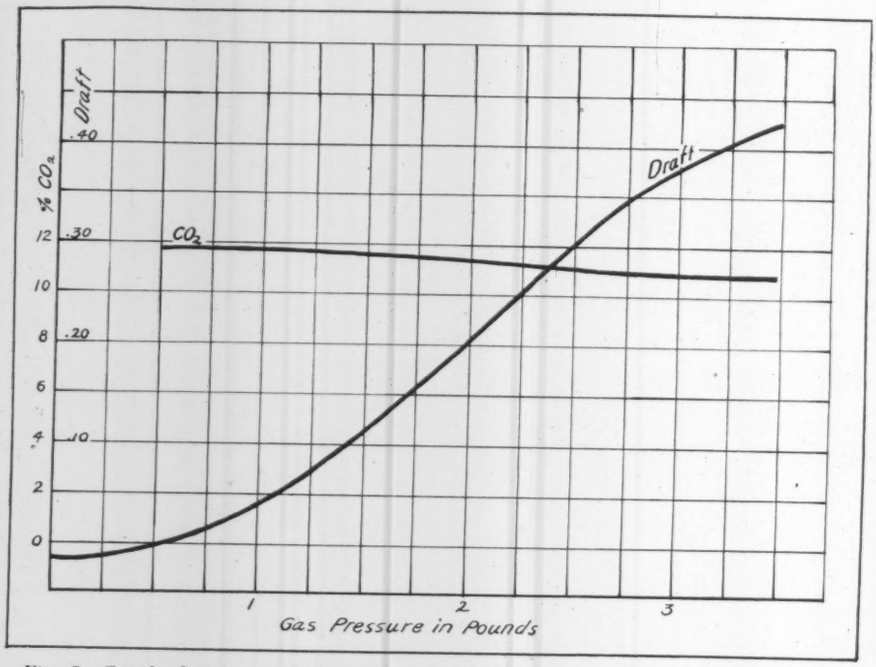



Fig. 7—Graph showing fuel air ratios to maintain maximum $CO_2$ in 300 hp boiler

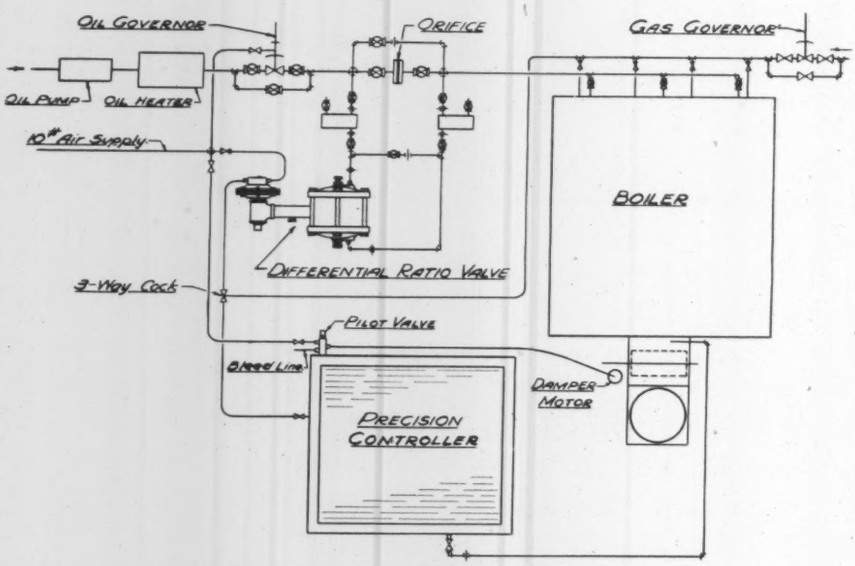



Fig. 8—Diagram showing installation of fuel air ratio controller

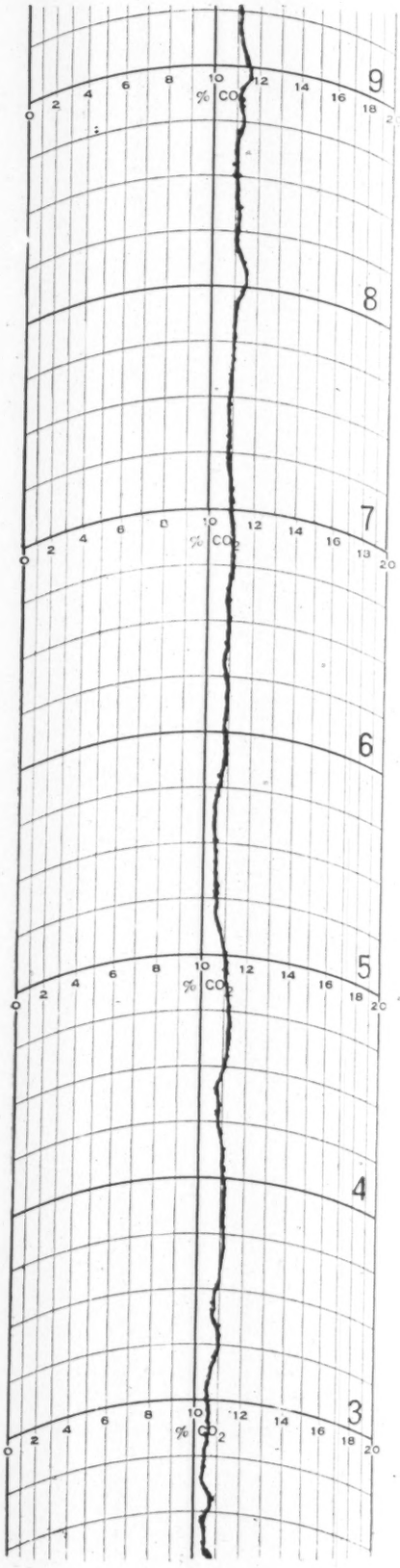
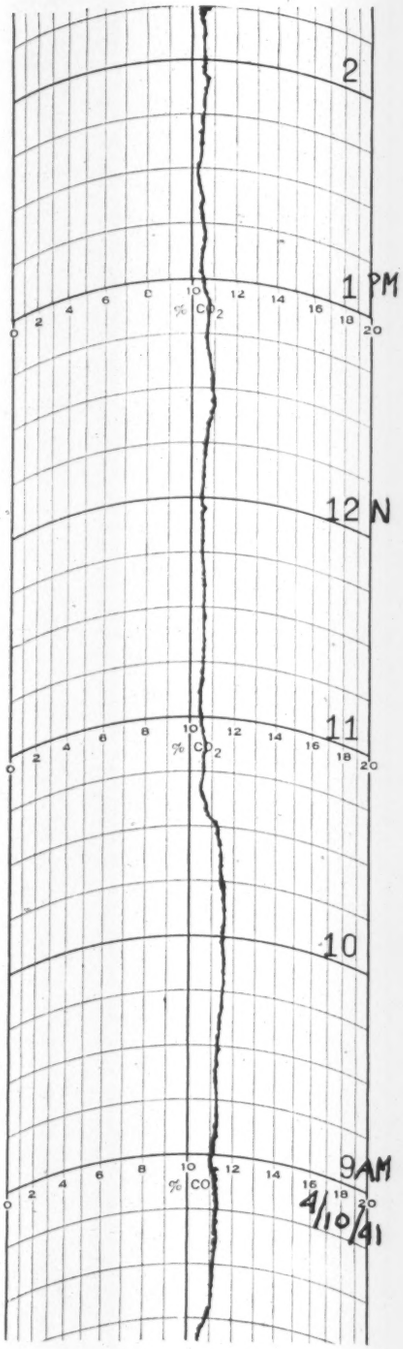
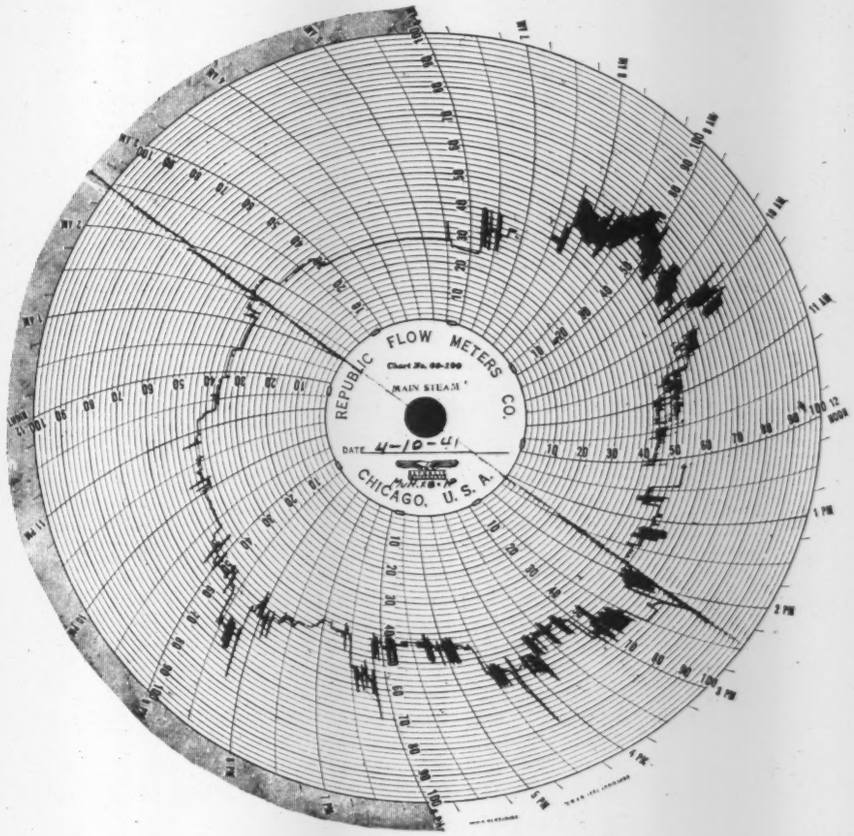



Fig. 9 (*Above and on opposite page*)—Flow chart and ranarex $CO_2$ recorder strip from an installation using fuel air ratio control




Test No. 1 of Table 2 that no stack control was provided and the excess air was 92 per cent with only the small burner bank operating. Test No. 2 of the same figure indicates that with the use of draft control excess air was 42 per cent. The problem created by this method of firing is strictly one of proper draft regulation, and to secure desirable results under the normal operating load requires proper draft adjustment at that rate. A method preferable to bank or sectional operation is a step control of the entire burner assembly which affords more accurate control of air, provided the burners possess suitable turn-down characteristics.

The application of modulating controls, of which Fig. 6 is typical, is suitable for both heating and power boilers which carry continuous modulating loads. It is also used more generally for horizontally fired units. The modulating controls employ a steam actuated throttling control valve, and either incorporated in it, or functioning with it, some mechanism capable of operating louvers and dampers. This arrangement offers a semblance of fuel air ratio control. Table 3 shows results derived from a control of this type.

Greater refinements in fuel air ratio controls are justified by the fuel saving in larger boilers requiring modulating control. The modulating control already described, and others of its type produce a straight line characteristic of fuel to air. Theoretically this is correct in that the ratio of fuel to air should remain proportional at all loads for a gas of any given composition. Practically, however, this is not true in boiler application for the reason that excess air is always required beyond the theoretical value to insure complete combustion. It is necessary to increase the percentage of excess air as the upper limit of furnace capacity is reached. Fig. 7 shows the deviations that occurred in one example and it will be found to be characteristic of all boilers to a greater or lesser degree. The variations which occur in stack draft will also affect the shape of the curve. Stack drafts are not constant at any given time for a given firing rate. Therefore to secure proper fuel air ratios at all rates of firing and under all draft conditions requires a mechanism which will automatically compensate for any changes in either of these two conditions. In its simplest form it consists of a draft diaphragm which is balanced against a gas pressure mechanism, thereby balancing the forces of draft against fuel. A change in either one or both of these forces operates the pilot valve arrangement to admit or release air pressure on a damper diaphragm. This provides independent movement of the damper and any changes in draft due to changes in

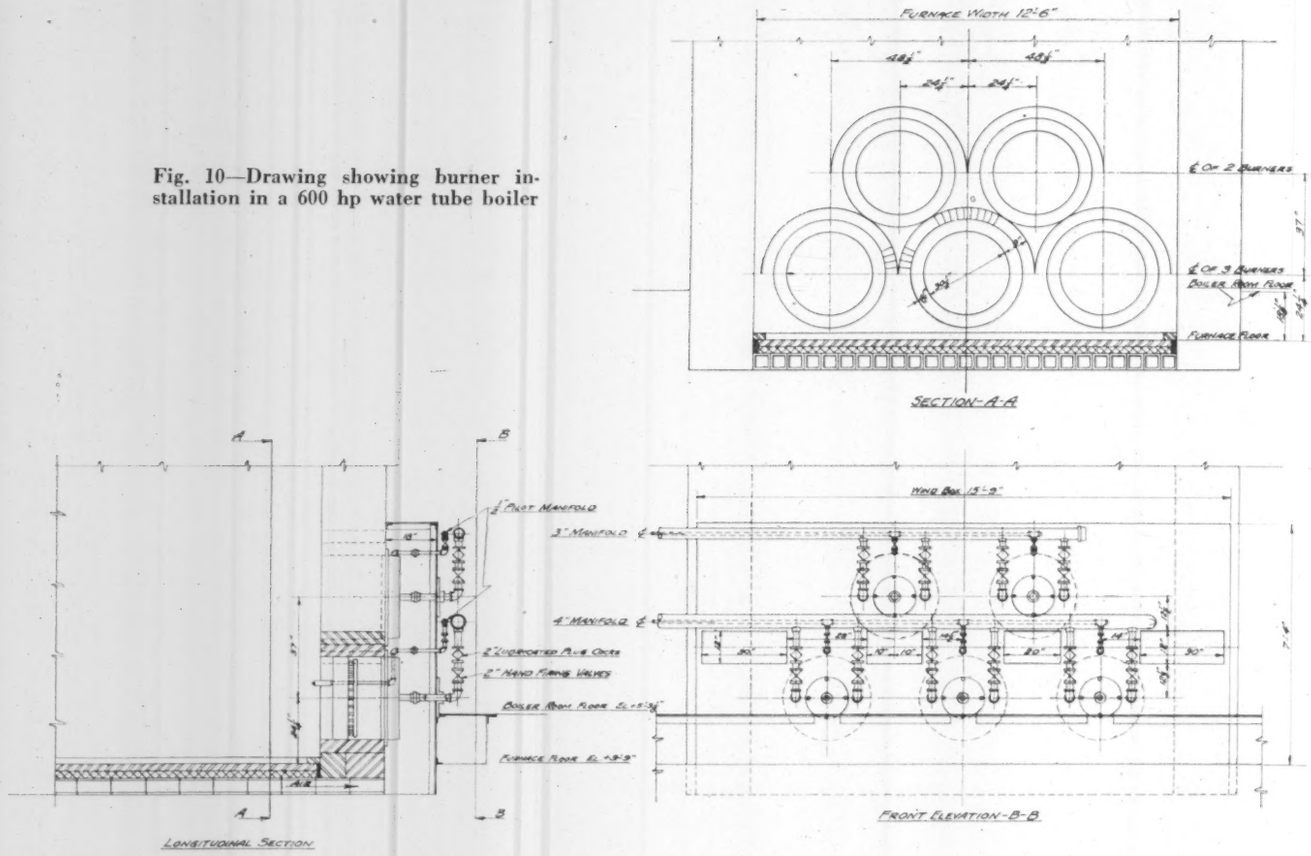



Fig. 10—Drawing showing burner installation in a 600 hp water tube boiler

stack effect, other than that for which the instrument is calibrated, will be corrected immediately by the automatic adjustment of the damper. Fig. 8 is the diagrammatic arrangement of a fuel air ratio control applied to a boiler. It will be noted that the measurement of air flow through the boiler, in other words, the draft loss through the boiler is taken as a measurement of air rate. The fuel pressure, controlled by the fuel governor, is taken as the measure of fuel rate. The air rate and fuel rate are balanced by the instrument, which is initially set by Orsat analysis. The quality of sensitive fuel air ratio control may be observed in Fig. 9 which indicates the $CO_2$ maintained with load fluctuations shown on the flow chart.

The reference to sectional operation of vertically fired units has its parallel in horizontally fired units. The requirement of burner capacity to satisfy maximum peak loads of short duration considerably in excess of normal operating demand justifies the application of this type of control. Fig. 10 shows the burner application in a 600 hp boiler. The maximum demand on this boiler is approximately 1500 hp and normal operation is less than nominal rating. The wide variation in load demands of this character is a severe measure of the flexibility of any burner or control equipment. Therefore the two upper burners are supplied from a gas line separate from the three lower burners. The registers of the two upper burners are controlled individually from those on the three lower burners, and the louvers in the windbox are likewise controlled separately.

The control system employed in this installation as shown in Fig. 11 is essentially the same as described in Fig. 8 with the following differences: The valves in the supply lines to both banks of burners are motor valves with gradual opening seats affording a parabolic flow characteristic. The motor valves are controlled by the valve positioner which varies the flow through them in direct proportion to the control pressure. This makes it possible to use any number of valves in similar manner, and exactly proportion the flow through them.

The master controller, which is sensitive to changes in steam pressure, transmits an impulse to the fuel control valve and to the air control simultaneously. The air control regulates the differential across the boiler, which produces a definite rate of flow for a given fuel demand by controlling dampers, louvers and air registers as required. A decrease in load beyond that which is desired to operate all burners, will be reflected in the gas pressure to the burners, and the pressure switch installed in the line to the three lower burners will close the fuel valve in the line to the two upper burners. Increasing demand will cause the valve to open again.

The measurement of regulation of the air supply is made between the windbox and furnace. When the gas is shut off from the two upper burners, thereby reducing the total burner orifice capacity by 40 per cent, the registers of these burners are closed to reduce the air orifice area by the same proportion. This permits the functioning of the ratio control without compensating adjustment.

The subject of automatic pilot control equipment is indeed a controversial one. Until recently the larger number of heating boilers have been equipped with plain pilots which have been developed to a

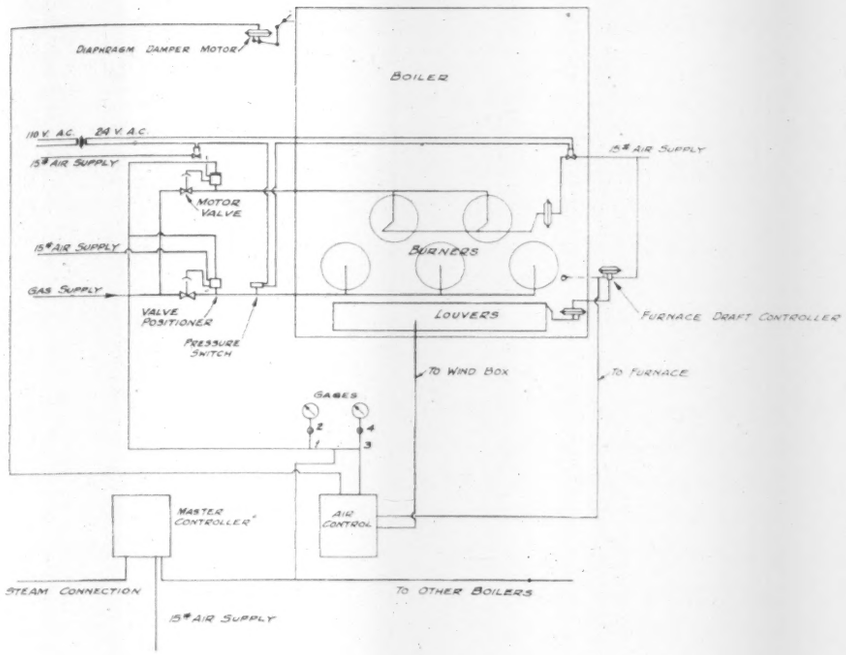



**Fig. 11—Drawing showing control system in a 600 hp water tube boiler**

reasonable point of stability for use with both vertically fired units and horizontally fired units.

The successful application of automatic pilot control may be considered only as a minimum safety requirement. As such, the automatic pilot control must be capable of igniting the entire burner assembly without any assistance of plain gas pilots, and it must be sufficiently rugged to withstand furnace conditions imposed upon it in heating boilers. It should also in the event of failure of the automatic itself, fail safely. There has been a reticence to endorse the use of any particular type of pilot, probably for the reason that the past recalls experiences of pilot failures. However in light of recent developments more thought should be given to their use. There is a growing tendency to adopt automatic pilots which operate on the principle of flame conductance on the larger heating as well as power generating boilers. Cost has retarded the acceptance of these types, but it should be placed above all else that positive ignition failure protection may only be evaluated in terms of resulting disaster, which in any case would be considerably greater than the cost of the most perfect control. Therefore in the application of any fuel, automatic pilot control systems, equal to the function they must serve, must be judged on the protection they afford and not the initial cost.

The general rule observed in all cases, regardless of whether automatic control is used, is to select plain gas pilots which are stable and durable in boiler furnaces, and to place a sufficient number of them in a manner to provide rapid and positive ignition of the burner assembly.

## *NORTH CAROLINA HOLDS SUMMER MEETING*

The North Carolina Chapter members held their summer meeting at the Berkshire Hotel, Pinehurst, N. C., July 18-20, and according to President Wallace, was well attended.

Among the members who attended with their wives were: Messrs. C. Z. Adams, P. L. Guest, H. H. Hill, F. E. Klages, Arvin Page, L. L. Vaughan, and W. M. Wallace, II. Other members present were: Messrs. E. R. Harding, Harry Hoffman and R. B. Rice. Also in attendance were Mr. and Mrs. A. H. Koch, who came up from Atlanta, Mr. and Mrs. Karl Selden and the guest speaker, Mr. Burch and his wife. Mr. Koch is the secretary of the Atlanta Chapter of the ASHVE.

Swimming was enjoyed on Friday afternoon, and in the evening the Board of Governors held a meeting at which Mr. Rice acted as proxy for the secretary, F. J. Reed, who is now in Washington with the Naval Ordnance Laboratory. Mr. Adams acted as temporary secretary.

***(Top)*** **North Carolina Chapter Presidents—Past and Present: R. B. Rice—1939, Arvin Page—1940, W. W. Wallace, II—1941**

***(Center)*** **Bridge Foursome at Pinehurst: Mmes. Selden, Hill, Wallace and Adams**

***(Bottom)*** **At Pinehurst Meeting: Messrs. Klages, Page, Harding and Wallace**

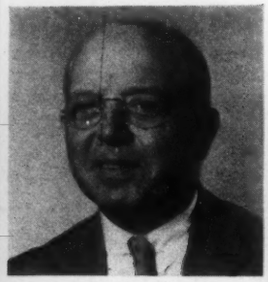
Saturday was devoted to golf and bridge, with a business session following luncheon. The banquet was held in the evening and Mr. Burch presented an interesting talk with illustrated colored slides showing the progress of the Great Smoky Mountain National Parkway, which connects with the Skyline Drive, originating in Virginia. Following Mr. Burch's talk, Mr. Rice presented a colored motion picture with sound effects showing many of the historical and scenic views of North Carolina. Mr. Klages acted as toastmaster, and Mr. Adams presented the prizes for golf, bridge and ping-pong.

The three-day visit of the North Carolina members to Pinehurst was enjoyed by all and the success of this meeting means that it will become a regular feature.

## Philadelphia

R. D. Touton
*Secretary*

H. H. Mather
*1st Vice-President*



H. Berkley Hedges
*President*

Edwin Elliott
*2nd Vice-President*

A. C. Caldwell
*Treasurer*

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C. B. Eastman
*Board of Governors*

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*1st Vice-President*

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*President*

C. F. Boester
*2nd Vice-President*

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E. R. Gauley
*Board of Governors*

# Philadelphia Invites You for '42

## 48th Annual Meeting and 7th International Exposition

**Committee on Arrangements**

M. F. Blankin, *General Chairman*

H. Berkley Hedges, *Vice-Chairman* — R. D. Touton, *Secretary* — C. B. Eastman, *Vice-Chairman*

**Honorary Committee**

*(Life Members)*

Homer Addams — J. D. Cassell
G. W. Barr — A. W. Luck
C. V. Haynes

*Hotels:* Karl Rugart, *Chairman*
*Publicity:* J. H. Hucker, *Chairman*
*Entertainment:* W. P. Culbert, *Chairman*
*Ladies:* Mrs. M. F. Blankin, *Chairman*
*Banquet:* H. H. Erickson, *Chairman*
*Transportation:* L. A. Childs, *Chairman*
*Inspections:* E. H. Dafter, *Chairman*
*Finance:* A. C. Caldwell, *Chairman*
*Sessions:* R. F. Hunger, *Chairman*
*Reception and Registration:* H. H. Mather, *Chairman*

MEMBERS and friends of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS are invited to spend the week of January 26, 1942 in Philadelphia when the Society holds its 48th Annual Meeting at the Bellevue-Stratford Hotel, the *National Warm Air Heating and Air Conditioning Association* convenes at the Benjamin Franklin and the 7th International Heating and Ventilating Exposition will be housed at the Commercial Museum.

A diversified program of technical sessions, entertainment and inspection trips has been planned by the Committee on Arrangements, of which M. F. Blankin is General Chairman and he extends a cordial invitation to all ASHVE members to enjoy the hospitality of Philadelphia in January. Headquarters for ASHVE meetings will be at the Bellevue-Stratford Hotel.

Philadelphia offers a variety of attractions and much of historic interest to the visitor. Since the Society's last visit a national shrine to Benjamin Franklin, the Society's patron saint, has been established in Philadelphia by Franklin Institute.

With the combined attractions of a varied technical program, an extensive exposition of modern equipment, and a series of attractive social functions, Philadelphia offers a week of enjoyable events in January.

Leading manufacturers of heating, ventilating and air conditioning equipment have already reserved 80 per cent of the available space in the 7th International Heating and Ventilating Exposition.

With progressive industrial expansion an outstanding characteristic of the current year, it has been forecast that the average level of business will reach the highest production point in one hundred years. This serves to emphasize some of the pressing engineering problems connected with comfort, efficiency and the health of the public which need prompt solution.

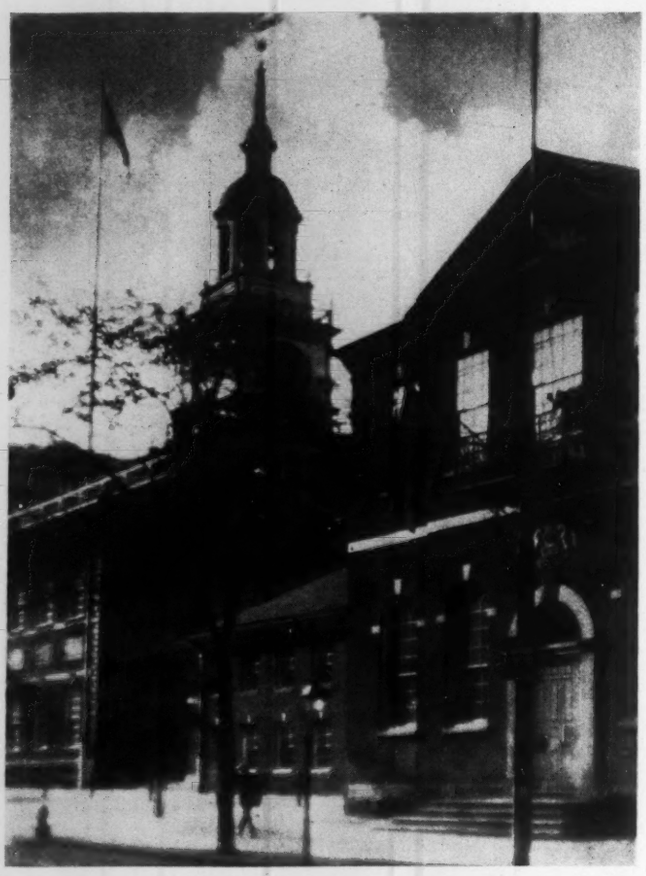
According to a statement by Walter L. Fleisher, President of ASHVE, recognition of the Society's leadership in research and its allied fields of activity has been given by government departments which require immediate answers to important defense problems. For these investigations, much of the research and capacity of our laboratories have been assigned.

Because heating, ventilating and air conditioning are essential industries, they enter into every industrial and public building program. For the army camp or home to maintain comfort, efficiency and the health of the occupants heating, ventilating and air conditioning must play an important part. For this reason, the exhibits of manufacturers at the 7th International Heating and Ventilating Exposition take on increasing importance.

The thinking of leaders in the field encourages the adoption of research results into practice as rapidly as possible. For that reason the

**Liberty Bell**

The Liberty Bell, which first proclaimed American Independence, is preserved in Independence Hall. It still hangs upon its original beam. It was originally cast in England and brought to Philadelphia in 1752 to be hung in the State House (Independence Hall) for the purpose of summoning the members of the Colonial Assembly of that time to its meetings. While being taken from the ship it was cracked and recast in 1755 in Philadelphia. Following the signing of the Declaration of Independence it was rung on every subsequent Fourth of July, until the year 1835, when it was cracked while being tolled for the burial of Chief Justice Marshall.



**Independence Hall**

Seventh International Heating & Ventilating Exposition, while presenting a complete panorama of the industry's progress, has the atmosphere of an engineering congress, rather than a merchandise display.

According to an announcement by Charles F. Roth, manager of the Exposition, President Fleisher has accepted the chairmanship of the Advisory Committee of the Exposition.

## *CHAPTER SPEAKERS FOR OCTOBER*

Plans prepared by the Local Chapters Committee, of which E. K. Campbell is Chairman, have been approved by the Council, and Local Chapters that desired the assistance of the Speakers Bureau will hear the subject of research discussed at their October Meetings.

The list of assignments announced by Mr. Campbell is as follows: *Dr. F. E. Giesecke*—October 7, Golden Gate, October 8, Southern California, and October 20, Oklahoma; *Prof. G. L. Larson*—October 6, Minnesota; *F. C. McIntosh*—October 13, Western New York, and October 14, Cincinnati; *E. P. Heckel*—October 13, Iowa, October 14, Nebraska, October 15, Oregon, October 16, Pacific Northwest, and October 21, Wisconsin; *Dr. F. C. Houghten*—October 9, Philadelphia, October 13, Illinois, October 14, Western Michigan, October 15, Northern Ohio, October 20, New York, and November 3, Ontario; *John James*—October 2, St. Louis, October 3, Kansas City, October 4, South Texas, October 6, North Texas, October 7, Delta, October 8, Atlanta, October 10, North Carolina, October 20, Montreal, October 21, Massachusetts, and October 22, Connecticut; *Prof. R. C. Jordan*—October 16, Manitoba.

**Exposition Advisory Committee**
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*Treasurer,* ASHVE

J. D. Cassell
*Life Member,* ASHVE

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*Past President,* ASHVE

C. V. Haynes
*Past President,* ASHVE

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*President,* Philadelphia Chapter of ASHVE

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York Ice Machinery Co.

Lee P. Hynes
Hynes Electric Heating Co.

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Consulting Engineer

F. A. Leser
*President,* Washington, D. C. Chapter of ASHVE

Thornton Lewis
*Past President,* ASHVE

F. D. Mensing
Consulting Engineer

L. E. Moody
Moody and Hutchison

Lee Nusbaum
Pennsylvania Engineering Co.

J. Earl Seiter
Consolidated Gas, Electric and Power Co.

T. H. Urdahl
Consulting Engineer

Warren Webster, Jr.
Warren Webster and Co.

C. F. Ames, Jr., *Chm. Ex. Com.*
*Insulation Board Institute*

Wharton Clay, *Secretary*
*National Mineral Wool Assn.*

Theo. Irving Coe, *Tech. Sec'y.*
*American Institute of Architects*

G. C. Diehl, *Council Member*
*American Institute of Consulting Engrs.*

W. R. Hainsworth, *1st Vice-Pres.*
*American Society of Refrigerating Engrs.*

W. B. Henderson, *Ex. Vice-Pres.*
*Air Conditioning and Refrigerating Machinery Assn.*

W. A. Herr, *Ex. Com.*
*National District Heating Assn.*

Frank Hoke, *President*
*Stoker Manufacturers Assn.*

R. A. Locke, *Mgr.*
*Steel Heating Boiler Institute*

A. F. Nass, *President*
*HPAC Contractors National Assn.*

A. J. Nesbitt, *Repr.*
*Industrial Unit Heater Assn.*

H. S. Sharp, *1st Vice-Pres.*
*National Warm Air Heating and Air Conditioning Assn.*

Ernest Szekely, *Vice-Pres.*
*Nat'l Assn. of Fan Manufacturers*

R. S. Waters, *Chairman*
*Institute of Boiler and Radiator Manufacturers*

E. C. Wente, *President*
*Acoustical Society of America*

John W. West, Jr., *Sec'y.*, Residential Sec.
*American Gas Association*

Ray G. Whipple, *President*
*Oil Burner Institute*

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R. F. Connell

A. E. Knibb

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S. S. Sanford

G. H. Tuttle

## Manitoba

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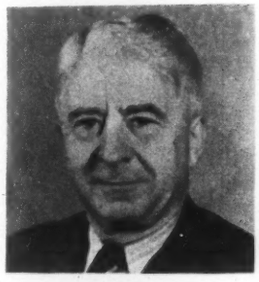
*President*



P. L. Charles
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F. L. Chester
*Secretary-Treasurer*

G. C. Davis
*Board of Governors*

William Worton
*Board of Governors*

Ivan McDonald
*Vice-President*

H. J. Metzger
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T. D. Stafford
*Board of Governors*

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C. H. Pesterfield
*Board of Governors*

# 1941-42 Chapter Officers

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E. W. Simons

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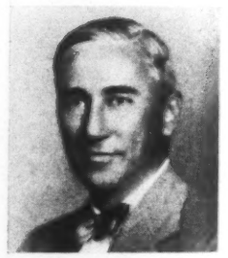
C. Z. Adams
*Board of Governors*

H. H. Hill
*Vice-President*

L. L. Vaughan
*Board of Governors*

F. J. Reed
*Secretary-Treasurer*



Arvin Page
*Board of Governors*

R. E. Le Riche
*Secretary*

C. W. May
*Board of Governors*

## Pacific Northwest

H. T. Griffith
*Vice-President*

M. J. Hauan
*Board of Governors*

F. J. Pratt
*President*

R. E. Chase
*Treasurer*

R. O. Wesley
*Board of Governors*

# NOMINATIONS FOR 1942

## Nominations for Officers and Council

The Nominating Committee appointed to select candidates for Officers of the Society for the coming year, 1942, submits the following list of nominees:

*For President:*

E. O. Eastwood, Seattle, Wash.

*For First Vice-President:*

M. F. Blankin, Philadelphia, Pa.

*For Second Vice-President:*

S. H. Downs, Kalamazoo, Mich.

*For Treasurer:*

E. K. Campbell, Kansas City, Mo.

*For Members of the Council:*

*Three-Year Term*

L. G. Miller, East Lansing, Mich.

A. J. Offner, New York, N. Y.

A. E. Stacey, Jr., New York, N. Y.

B. M. Woods, Berkeley, Calif.

Respectfully submitted,
*NOMINATING COMMITTEE,*
F. C. McIntosh, *Chairman.*

In accordance with the provisions of the Society's Constitution, By-Laws and Rules, ballots containing the names of the above candidates will be sent to the membership for voting upon prior to the Annual Meeting in January.

*Art. B-VIII—Section* 11. The Nominating Committee shall consist of one (1) member eligible to vote designated by each Chapter, or his alternate also appointed by the Chapter. The Secretary of each Chapter shall certify to the Secretary of the Society on or before January first the names of the member and alternate selected.

The Committee shall meet at the Annual Meeting of the Society at the call of the Secretary of the Society and shall effect its own organization and elect its own Chairman. At the Semi-Annual Meeting of the Society, if possible, the Nominating Committee shall select the nominees for the ensuing year for the offices of President, First Vice-President, Second Vice-President, Treasurer, and four (4) members of the Council. In any event the names of the nominees shall be certified to the Secretary of the Society before September twentieth, with the written consent of each nominee to fill the office for which he has been selected and their names with the offices to which they have been nominated shall be published in the October issue of the Journal.

*Art. B-IX—Section* 2. The Secretary shall prepare ballots with the names of all candidates and forward them to the members, eligible to vote, at least thirty (30) days before the date of the Annual Meeting.

## Nominations for Members of Committee on Research

Although committees of the Society are usually appointed, in view of the great importance of the Committee on Research and the financial responsibility it would be called upon to assume, it has been made more representative of the entire membership of the Society by a process of *election*. The election is governed by the By-Laws for the election of officers, with the single exception that Members of the Committee on Research are nominated by the Council instead of by a Nominating Committee.

In accordance with the Regulations for the Government of the Research Laboratory, the Council announces the nomination of the following members of the Committee for election to succeed those members whose present terms expire January, 1942:

*Three-Year Term*

C. F. Boester, St. Louis, Mo.

J. A. Goff, Philadelphia, Pa.

W. E. Heibel, New York, N. Y.

C.-E. A. Winslow, New Haven, Conn.

B. M. Woods, Berkeley, Calif.

The regulations governing the nomination and election of members of the Committee on Research are as follows:

Article II—Organization

*Section* 1. *Committee on Research*—There shall be a standing committee known as the Committee on Research, consisting of fifteen (15) members each serving for three (3) years and five (5) retiring each year. The outgoing Chairman if serving in that capacity during the last year of his three (3) year term on the Committee shall without election become an additional member of the Committee on Research for one (1) year.

(*a*) The Council shall nominate previous to July first of each year five (5) members to fill the vacancies of those retiring at the next Annual Meeting.

(*b*) The nominations made by the Council shall be published in the October issue of the Society's Journal.

(*c*) Any ten (10) members of the Society eligible to vote may present to the Secretary over their signatures, the name of one (1) or more additional nominees for the Committee on Research, provided such name or names are presented at least sixty (60) days prior to the next Annual Meeting, and such additional nominations shall be placed on the ballot opposite the nominations made by the Council.

(*d*) The election shall otherwise conform to the regulations provided for the election of officers of the Society in the Constitution, By-Laws and Rules.

(*e*) Vacancies may be filled by the Council, such persons chosen by the Council to serve until a successor is elected at the next Annual Meeting.